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DESIGN · PRODUCTION · MATERIALS

Vol. 51 No. 5 MAY 1961 PRICE: 3s. 6d.

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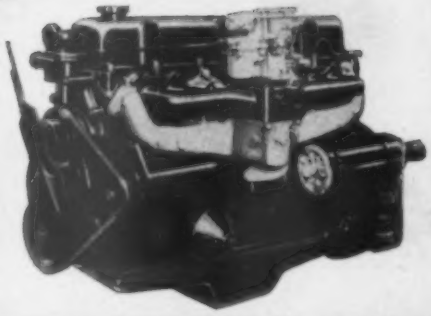

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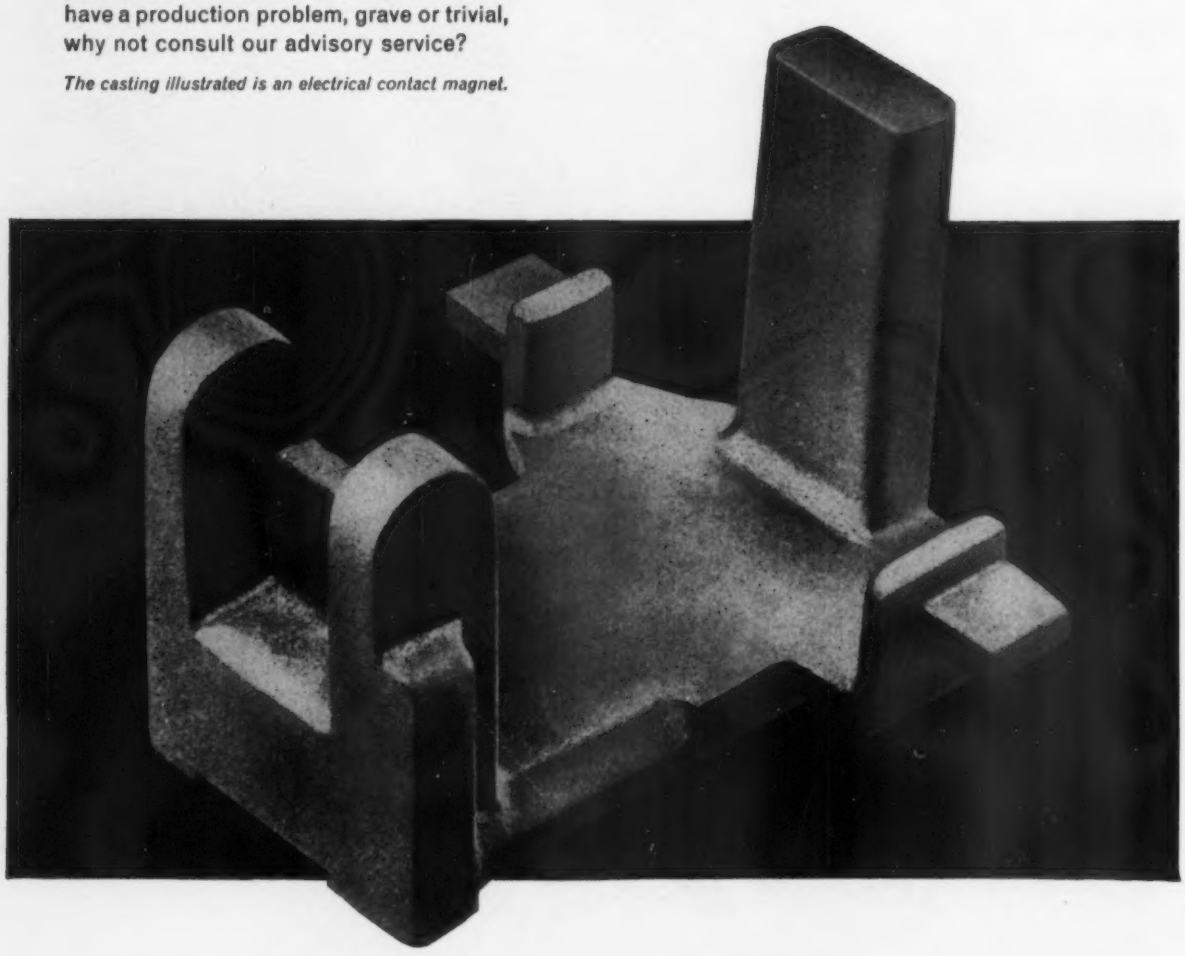

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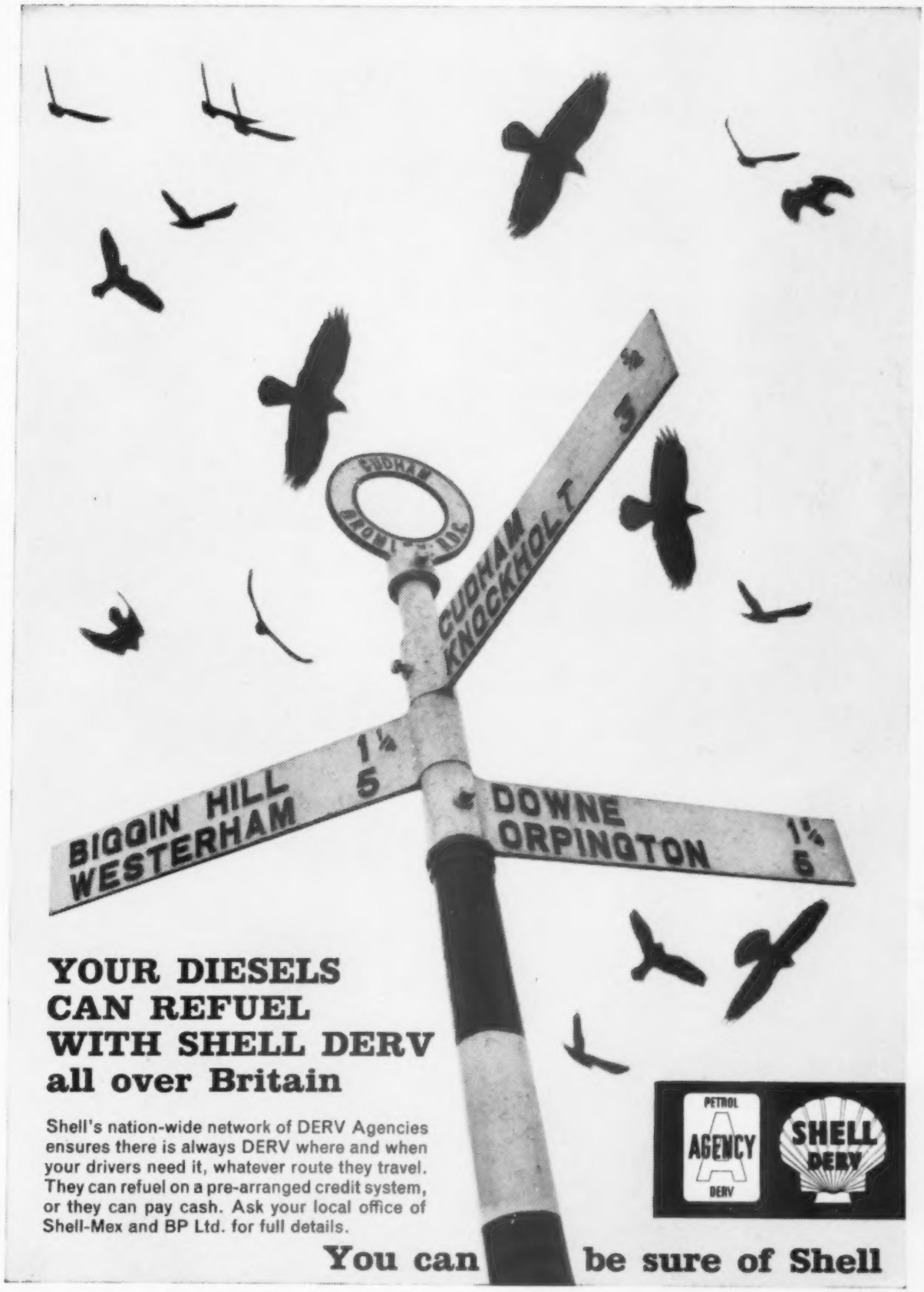
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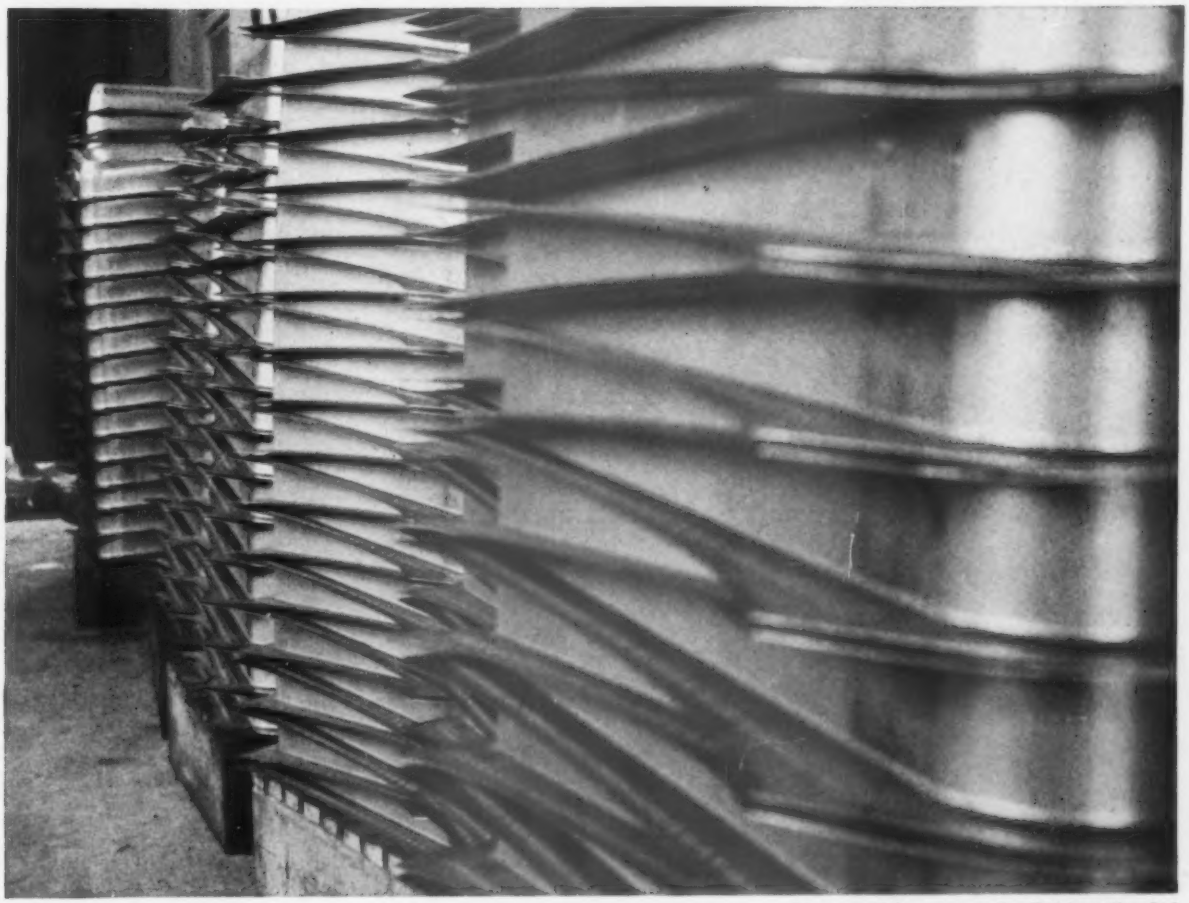

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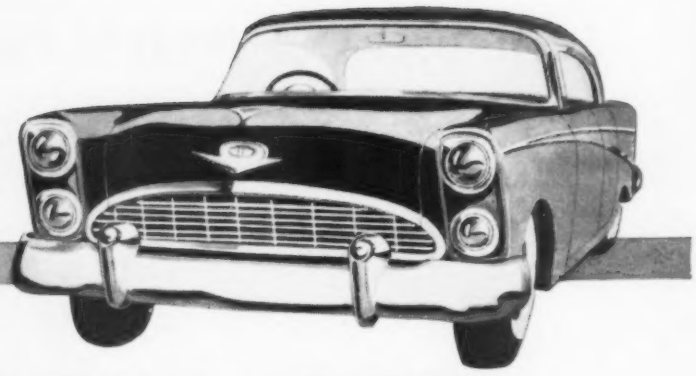

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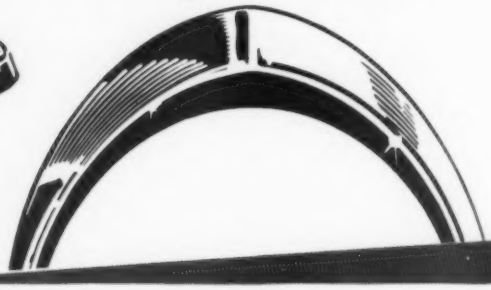

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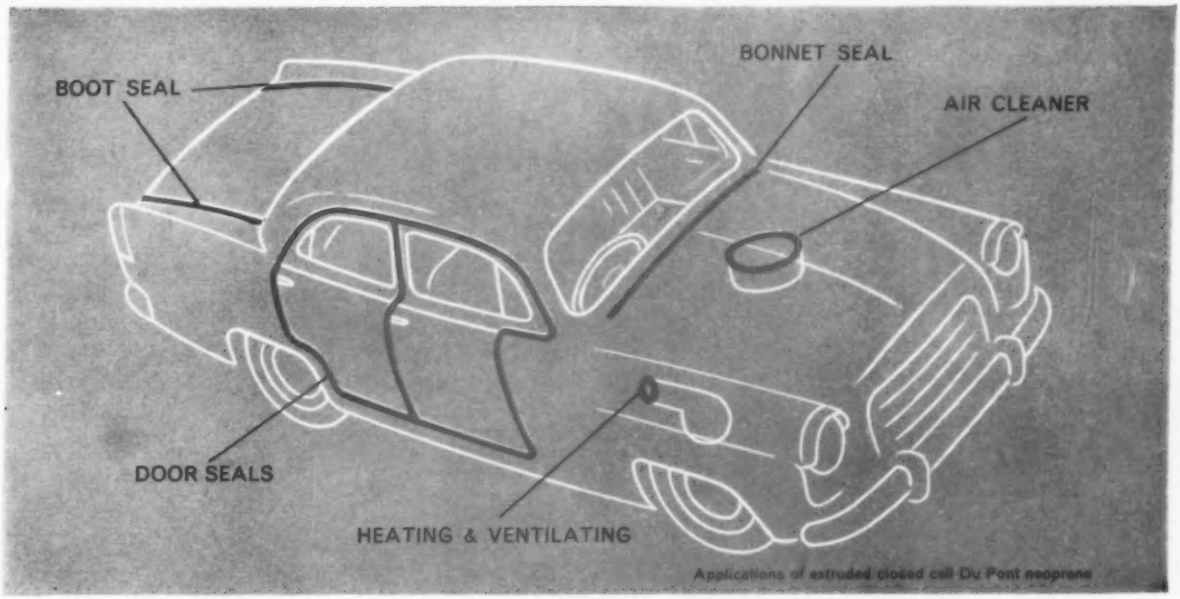
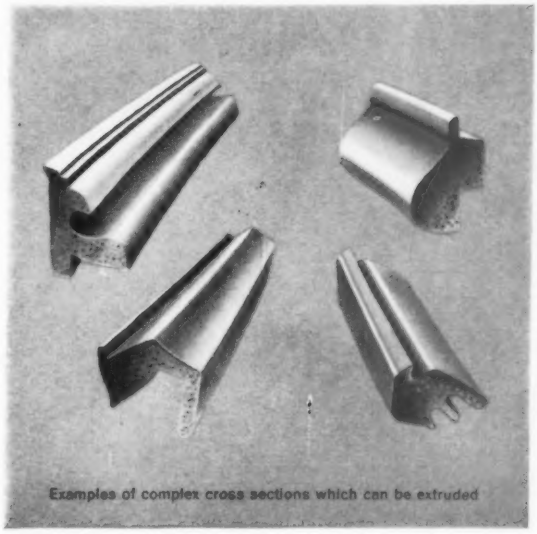


Applications of extruded closed cell Du Pont neoprene


Examples of complex cross sections which can be extruded

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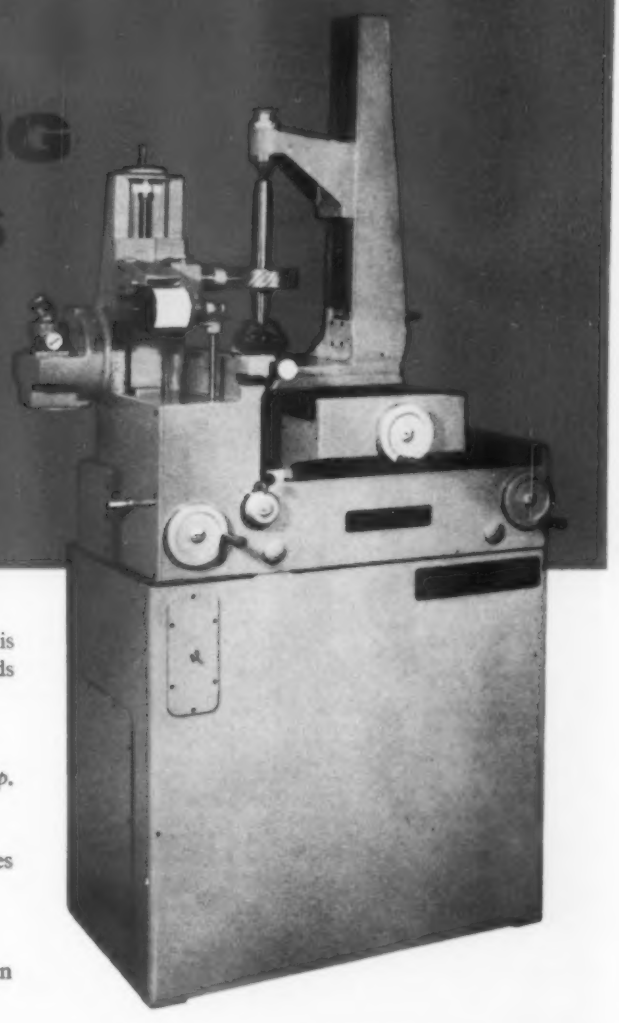

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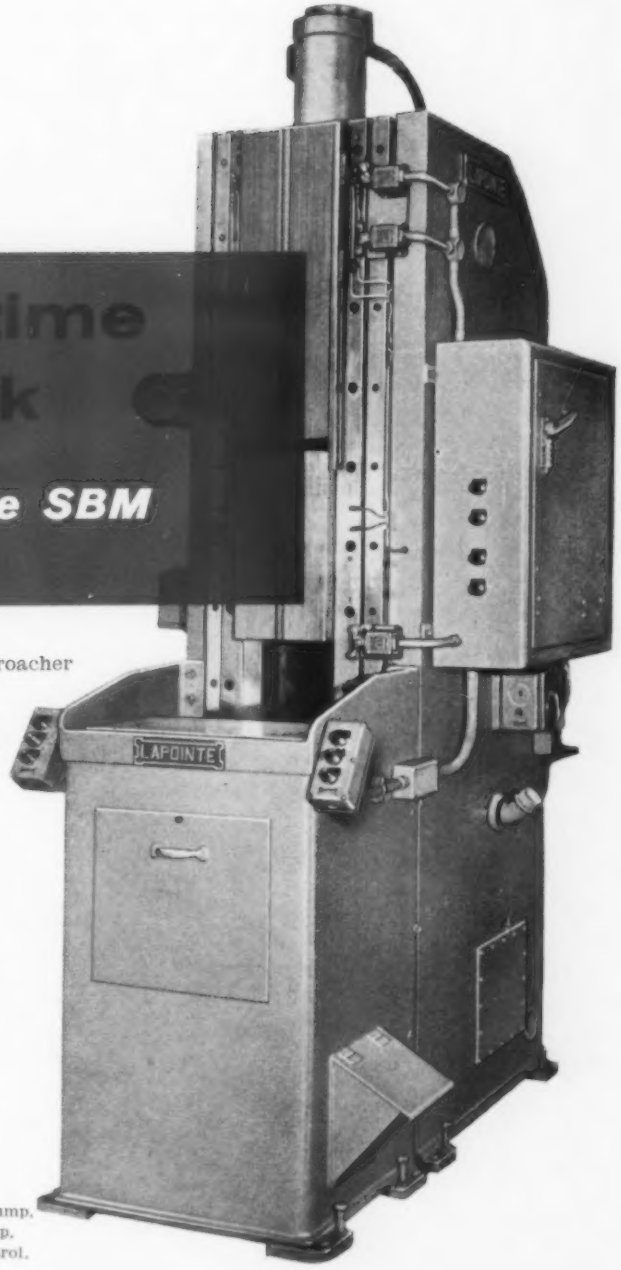

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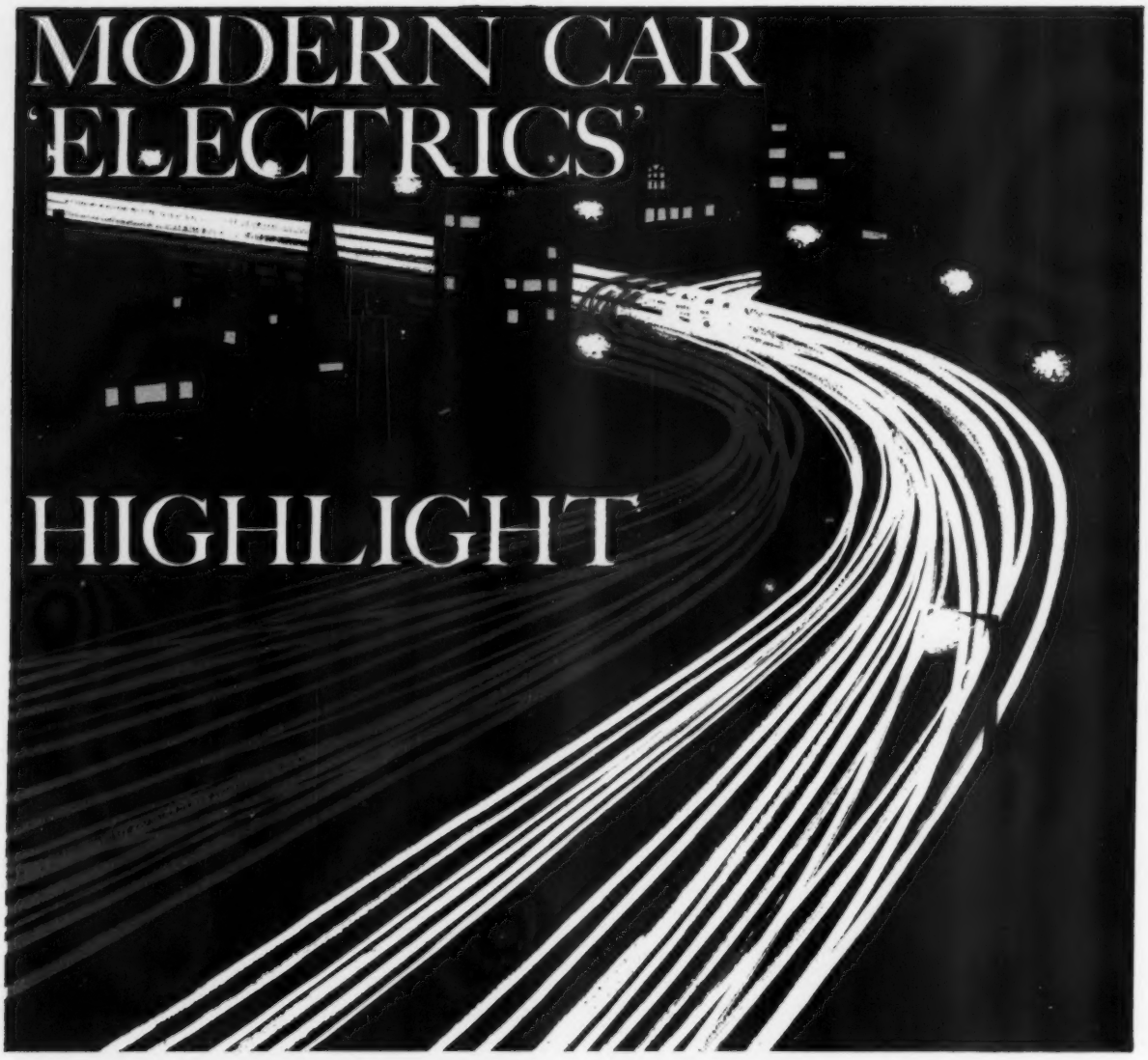

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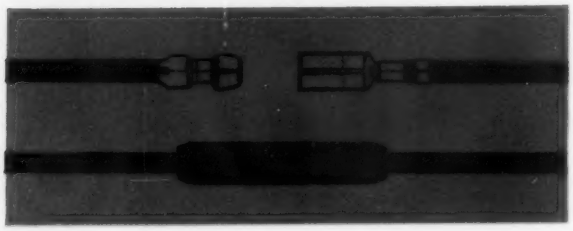

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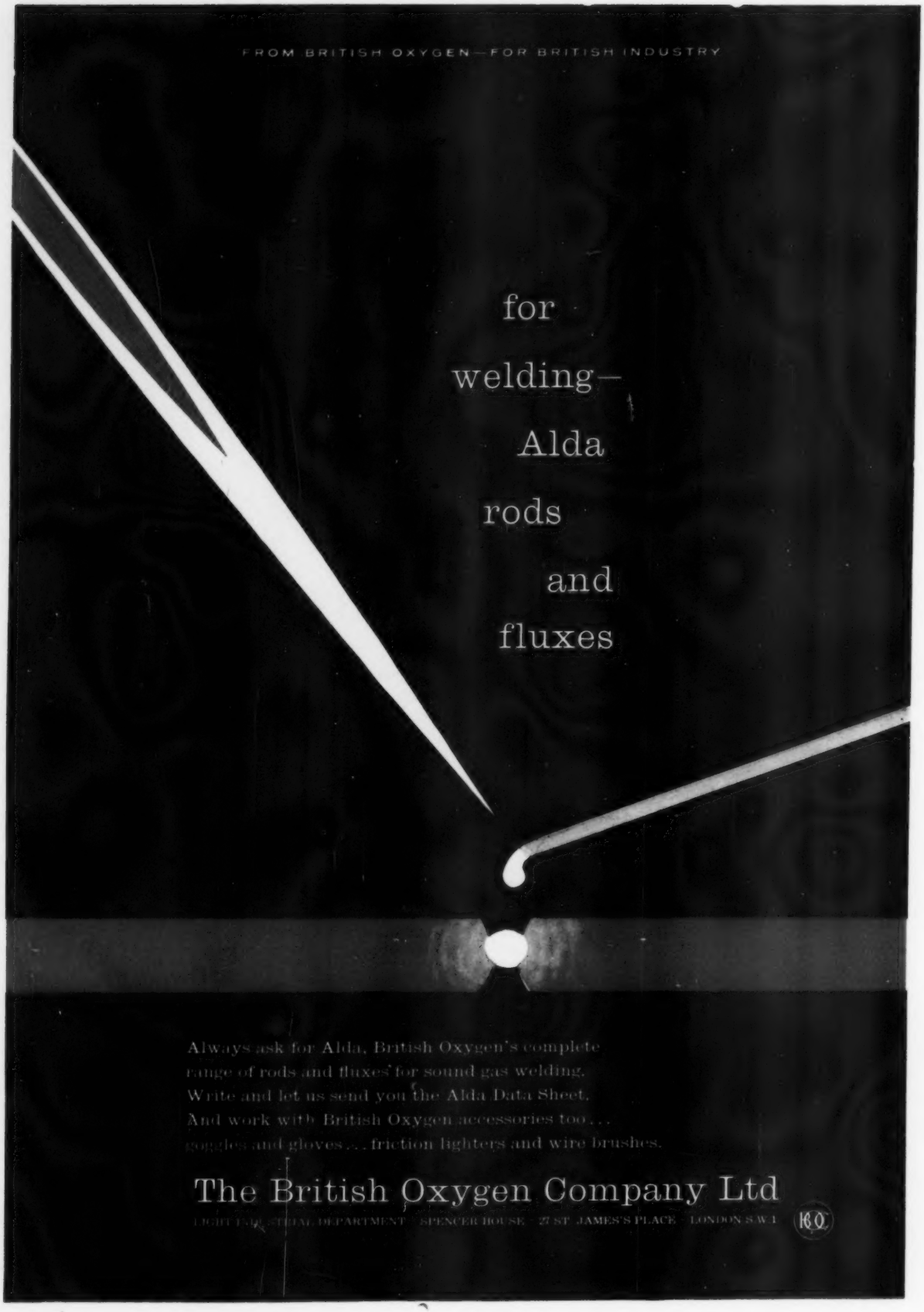
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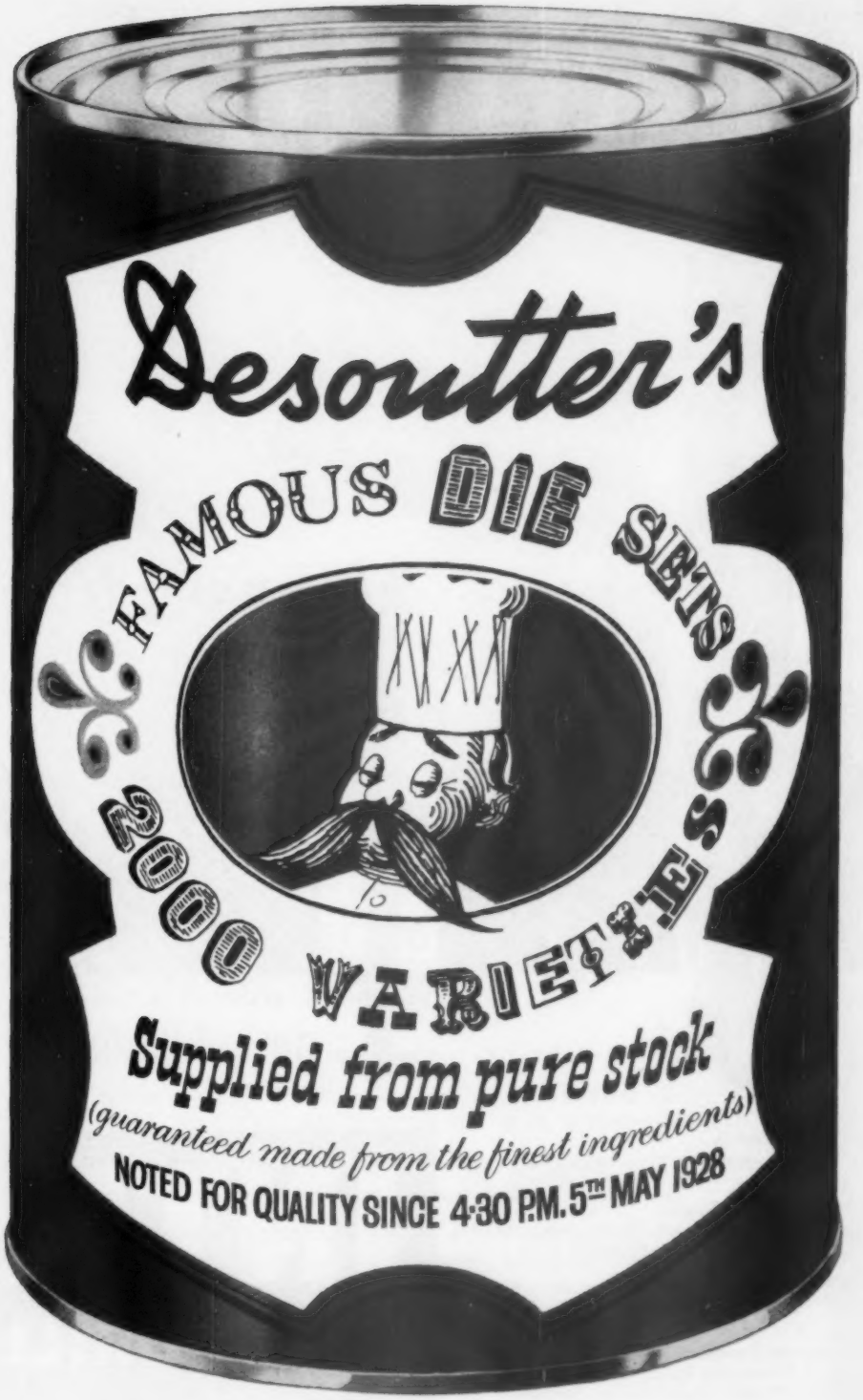
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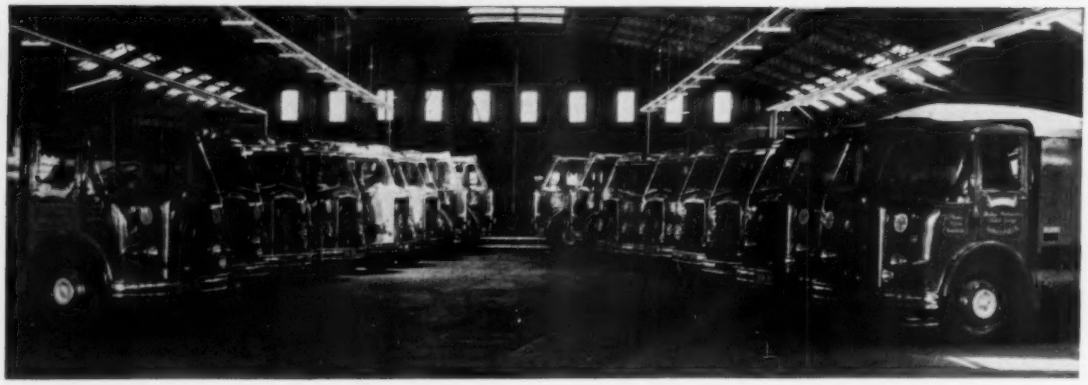


Part of the fleet of Hanson vehicles.

That the Huckbolt high-speed fastening system really does cut production costs was conclusively proved by The Service Garage (Brighouse) Ltd. and Comberhill Wakefield Garages Ltd. during the building of special light alloy bodies for Messrs. R. Hanson & Son Ltd., bulk hauliers of Wakefield, Yorkshire.

Designed for transportation of bulk materials as an alternative to rail transport, each vehicle uses 600 Huckbolts to provide the strength and rigidity necessary for this gruelling service.

In addition to the saving in time and labour cost, a further special advantage of the Huckbolt system on the contract was the quietness of operation — previously, weekend working had been impossible with normal riveting because of the annoyance to neighbours.

THE HIGH SPEED FASTENING SYSTEM THAT OUTDATES NUTS, BOLTS AND SOLID RIVETS

Huckbolts are superseding conventional fastening methods for many kinds of assembly operations. This is because of their speed of application (up to 20 a minute in many cases), their high positive clench and sealing qualities, and their resistance to vibration. Tensioning of the Huckbolt is completely mechanical; unskilled labour can therefore use the system with the certainty of efficient fastening. *Ask for more details.*

Close-up showing Huckbolts being applied. No skilled welders needed. No skilled riveters needed.

AVDEL INDUSTRIAL FASTENERS

AVDEL LIMITED
WELWYN GARDEN CITY, HERTS.
TELEPHONE: WELWYN GARDEN 28161.

*Manufactured under licence from the Huck Manufacturing Co. of Detroit, U.S.A. Huck Patents 629098, 766137, 790785.

AD 49

METALASTIK CONTRASONIC SPRING SHACKLES

PATENT APPLIED FOR

Isolating road noise

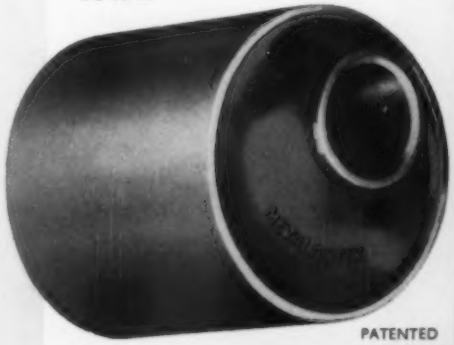
It is the *volume* of rubber that is the determining factor in isolating road noise.

Here are two very successful Metalastik components; the Contrasonic spring shackle, evolved for the 3-litre Rover and the Metaxentric bush. The construction of the Contrasonic shackle is self-evident, also the canting of the twin units to resist lateral stresses. Fore-and-aft movement, as the leaf spring changes in length, is accommodated by the rubber in shear; weight by combined compression and shear.

A Metaxentric bush in the front eye of the spring and a Contrasonic shackle at the rear provides the most effective barrier yet devised against the transmission and amplification of noise.

METAXENTRIC BUSHES

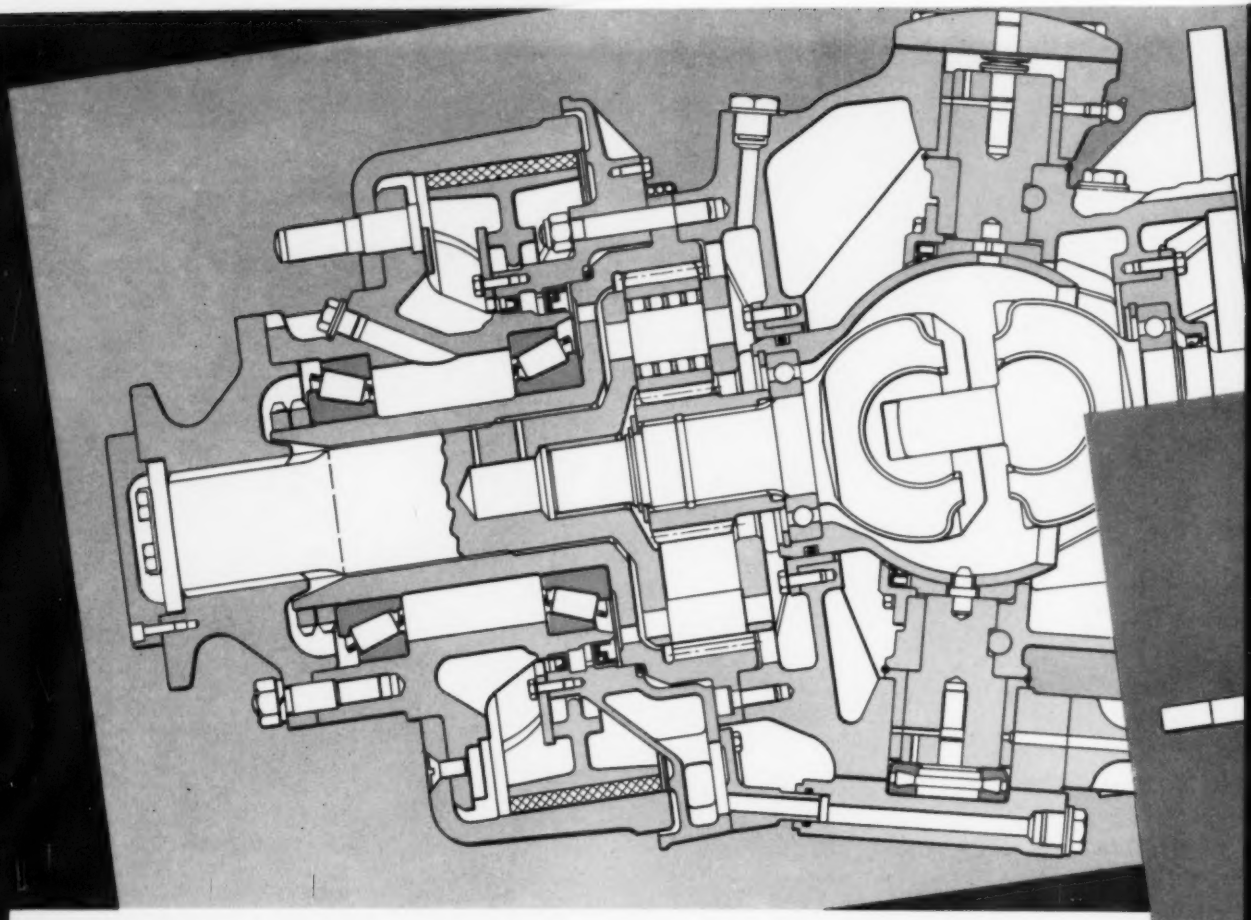
First fitted in spring eyes by the Standard Motor Company, Metaxentric bushes give much higher vertical deflections than concentric types. In fore-and-aft directions they are stiffer than in the vertical plane, the ratio being about 2·5 to 1.



PATENTED

METALASTIK

METALASTIK LTD., LEICESTER



The 'Gainsborough' medium wheel tractor

This robust and powerful machine, manufactured by Marshall, Sons & Co. Ltd., is driven and steered by all four wheels.

The drawings show some of the interesting features of the design. The six-cylinder diesel engine is at the back, and the drive is taken through a fluid flywheel and pre-selective planetary gearbox, thence through a right angle forward-and-reverse drive and spur reduction to the differential cage.

The differential-cum-reduction box, bottom right, divides the drive through final spur reduction gears which in turn drive Tracta universal joints within the steering heads, as shown top left. The drive passes to the front road wheels through planetary reduction gearing in the hubs.

The spur wheels (bottom right) carry bevel wheels which drive, at increased speed to reduce torque, the universal shafts passing along each side of the vehicle to the rear, where right-angle-drive reduction boxes (as top right) take the drive to rear steering heads, identical with the others. As will be seen, Timken bearings are employed extensively to meet the combined journal and thrust loads.

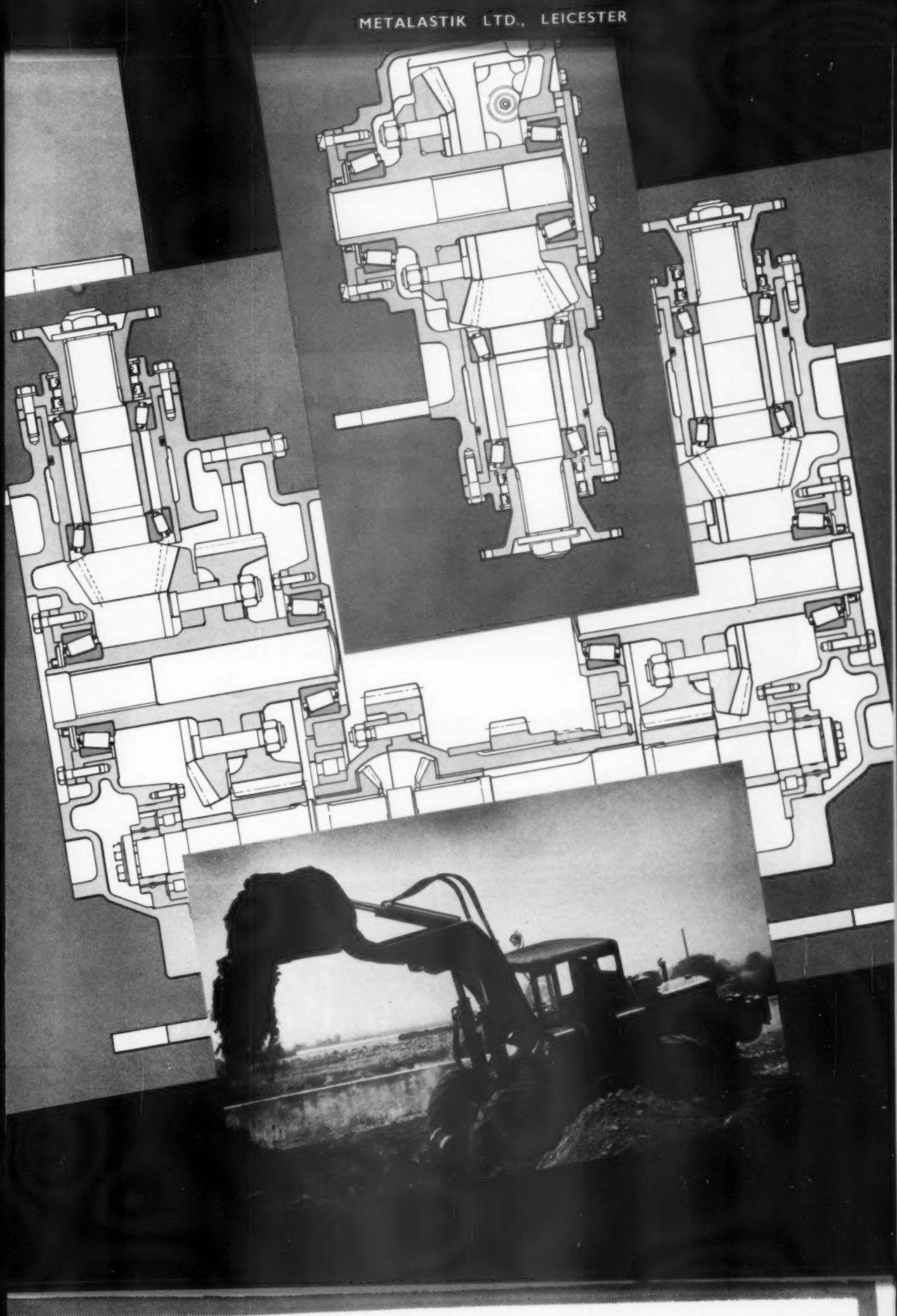
British Timken, Duston, Northampton, Division of The Timken Roller Bearing Company. Timken bearings manufactured in England, Australia, Brazil, Canada, France and U.S.A.

TIMKEN®

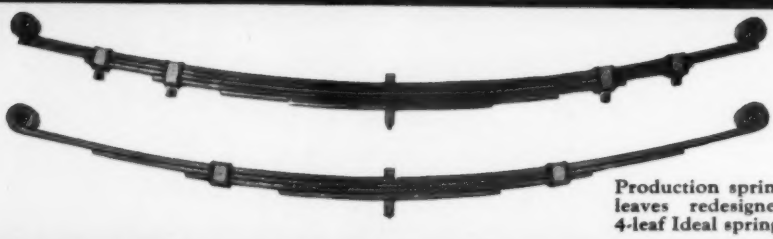
REGISTERED TRADE-MARK

tapered roller bearings

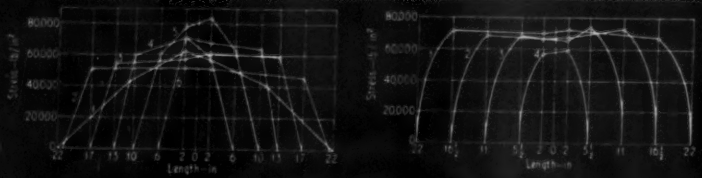
METALASTIK LTD., LEICESTER



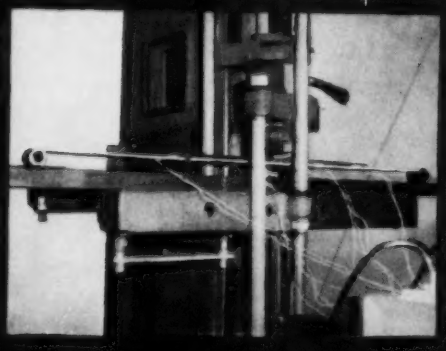
HOW TO SAVE SPRING WEIGHT WITHOUT INCREASING STRESS



Production spring with 6 leaves redesigned as a 4-leaf Ideal spring.



15-leaf commercial vehicle spring redesigned as 12-leaf Featherlight equivalent without increase in maximum stress.

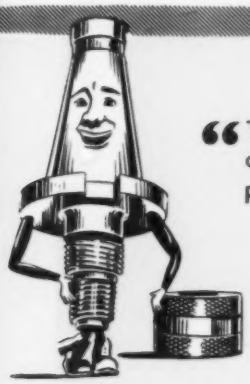


TWS
'featherlight'
IDEAL SPRING
[PATENTS PENDING]

TOLEDO WOODHEAD SPRINGS LIMITED
AYCLIFFE NR. DARLINGTON and SHEFFIELD 3

TWS 81

Screw-locked milling... AT ITS SIMPLEST & MOST EFFICIENT



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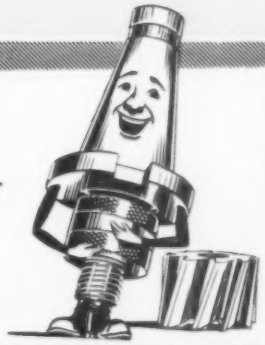
DORMER HELI-MATIC ARBORS AND SCREWED BORE CUTTERS

This high-efficiency combination simplifies screw-locked milling and achieves the greatest productivity.
Ease of assembly, precision in performance, and duration of working life, are the outstanding features of the Heli-Matic equipment.

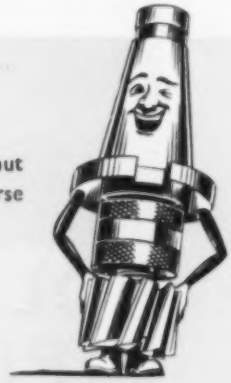
FIRST STAGE

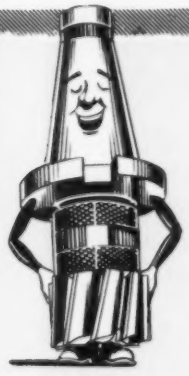
Screw on the nut by hand—as far as it will go.



SECOND STAGE

Screw on the cutter to meet the nut —then release by a part reverse turn.





THIRD STAGE

Screw back the nut firmly against the cutter by hand—immediately cutting starts the cutter will lock solidly against the nut, making a rigid tool assembly.

TO RELEASE

Apply the spanner to the nut and a few sharp hammer blows will release the nut from the cutter, which can then be screwed off by hand"

You can't go wrong—send for the Heli-Matic brochure giving the full range of Arbors and Cutters.

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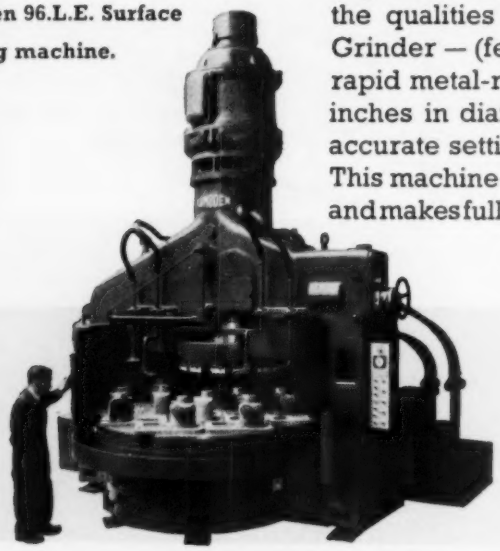
IN SURFACE GRINDING

Lumsden 96.L.E. Surface grinding machine.



High standard, accurate surface finishing are among the qualities attributed to the Lumsden 96.L.E. Surface Grinder — (featured below). This is a high production rapid metal-removing machine, with a work table of 100 inches in diameter and yet is capable of sensitive and accurate setting.

This machine is fitted throughout with Hoffmann bearings, and makes full use of their polished precision, dependability and quality, ensuring maximum efficiency and smoothness of operation at all times.

HOFFMANN
BALL AND ROLLER BEARINGS

Photograph by courtesy of the Lumsden Machine Company Limited of Gateshead

THE HOFFMANN MANUFACTURING CO. LTD., CHELMSFORD, ESSEX.

TELEPHONE: CHELMSFORD 3151 TELEX No. 1951.

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Using the principle of jigless boring, whereby work can be machined from two, three or four sides at one setting for milling, boring, drilling, tapping, facing or counter-boring operations in precise relationship, the Jigmil is the ideal machine for this type of work.

Two sizes of Jigmil are now manufactured in this country—the 2B-36 *(capacity* 24″ *vertical* × 36″ *horizontal travel and* 2½″ *dia. spindle) and the* 3H-48 *(capacity* 36″ *vertical* × 48″ *horizontal travel and* 3″ *dia. spindle). Both sizes can be adapted to suit the DATATROL dial and tape control system.*

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Great news for British manufacturers! Phoenix are now producing rubber to metal bonded anti-vibration units which are designed by Getefo, acknowledged leaders in anti-vibration engineering. Getefo, or to give them their full name, Gesellschaft Fur Technischen Fortschritt MBH of Koblenz, pioneered the study of the dynamics of anti-vibration problems. Now, in this new association with Phoenix, the knowledge gained in their researches is made available to British industry.

Measuring the internal temperature of the rubber of an anti-vibration mounting, during dynamic testing.

If you make a product which is subject to vibration, Phoenix can help you!

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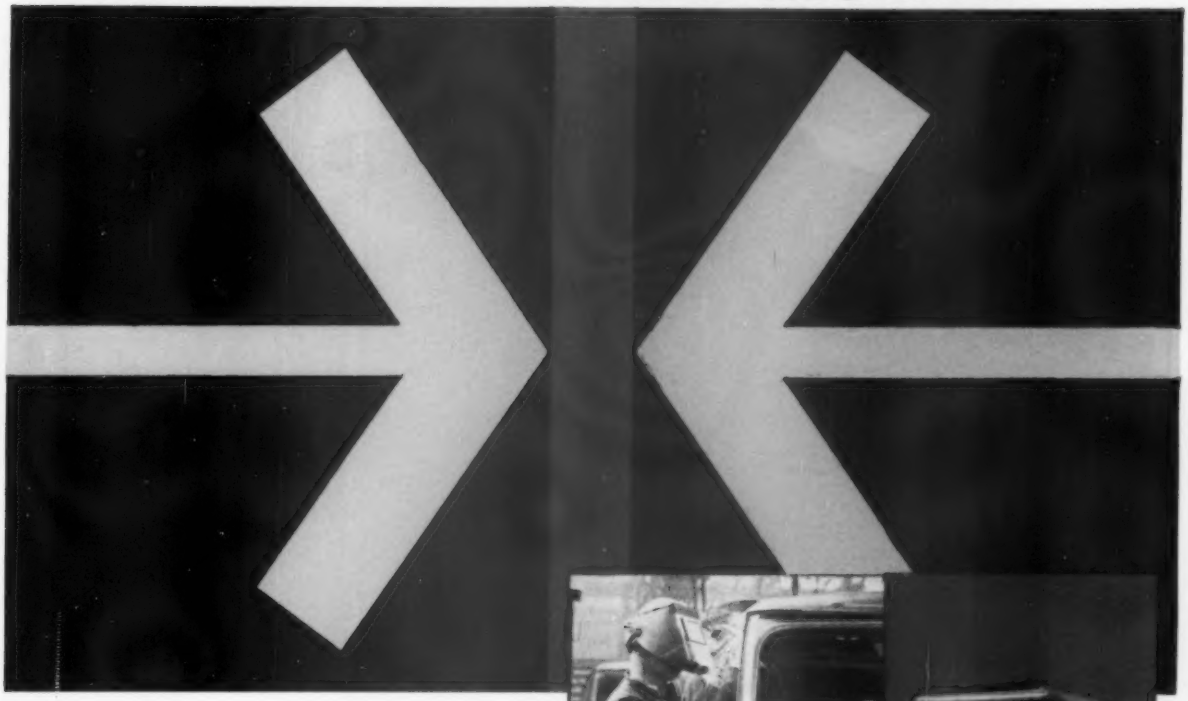

We don't suppose you weld across half-inch gaps in 20 swg mild steel sheet. The ability to produce flawless welds in such conditions is the measure of the efficiency of the BOC MRCS Slope Controlled Rectifier with the Lynx wire-feed equipment and ST2 torch.

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- *Easily portable Lynx wire-feed and gas-control equipment.*
- *ST2 Torch—featherweight, easy to handle even in awkward positions.*

Let us show you how well this equipment will do your particular work. Write to:

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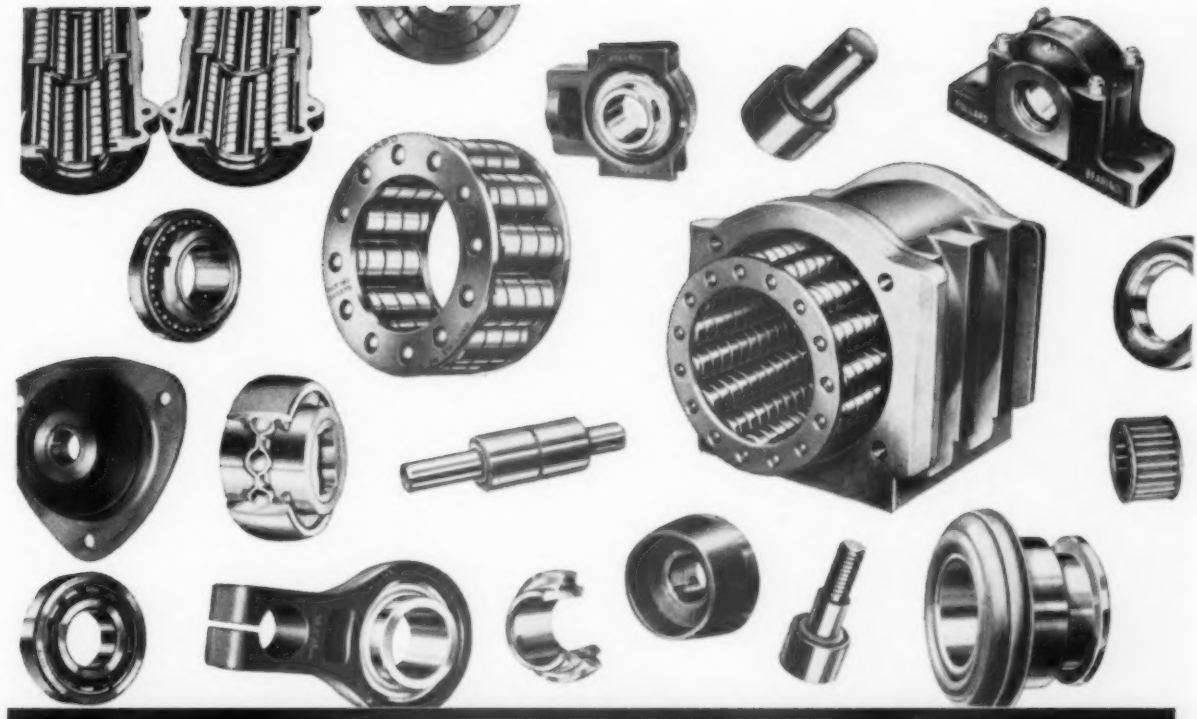

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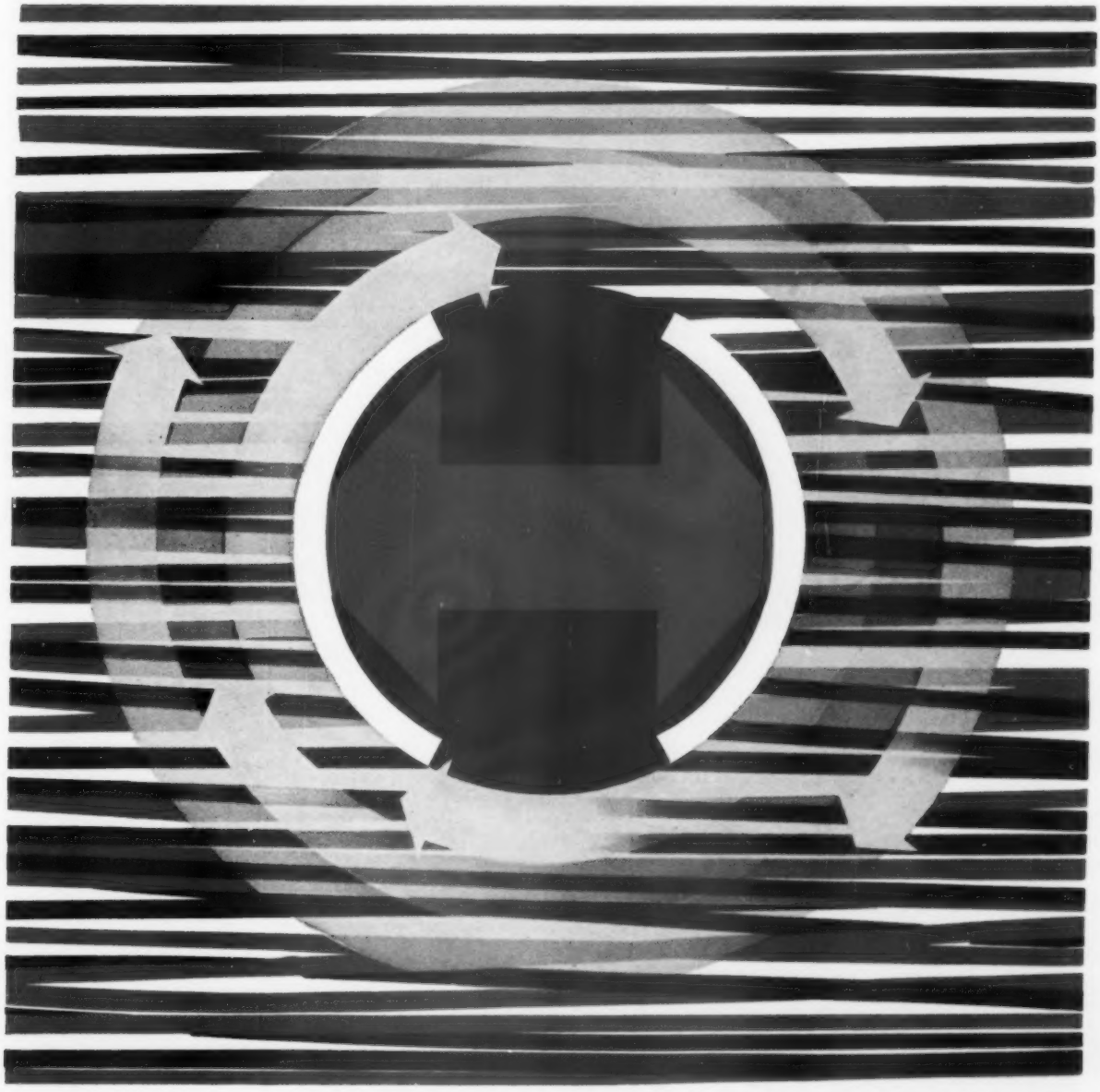

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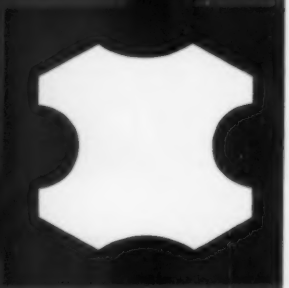

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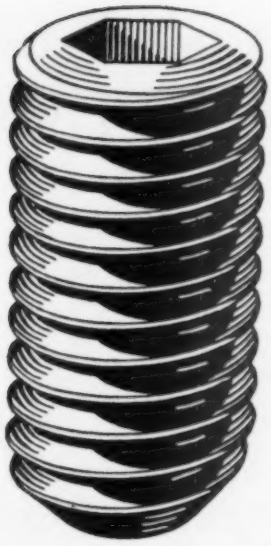

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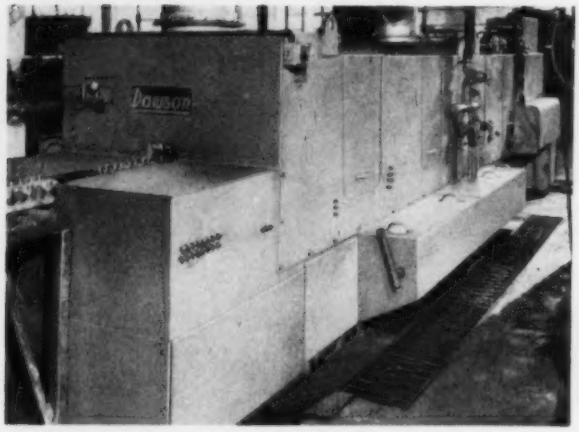
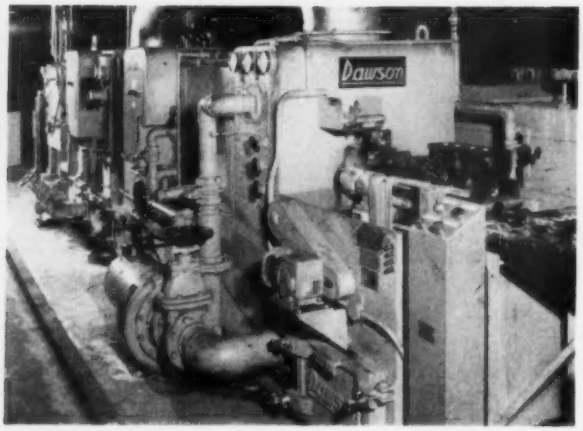
DAWSON experience is vital to satisfactory CLEANING & DEGREASING

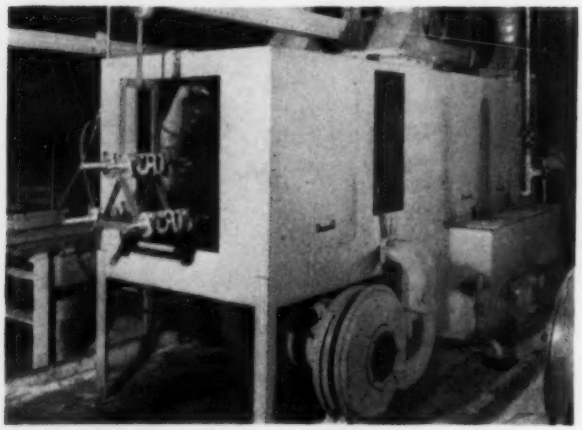
Advice based on experience ensures that Dawson put forward the right machine for the job. As can be seen from the illustrations, many machines have been supplied to the Motor Industry for high output, automatic cleaning and degreasing of a wide variety of components. Write today for a catalogue describing Dawson Equipment or ask for a specialist to discuss your particular requirements.



Dawson Machine for cleaning and degreasing Vauxhall cylinder heads.



Dawson Automatic Cylinder Block washing Machine showing an Austin Seven block about to enter the machine.



Dawson Automatic washing machine for Austin Seven Crankshafts, viewed from the exit end.

Dawson Type 'C' machine handling tractor components at the International Harvester Co. of Great Britain Ltd.

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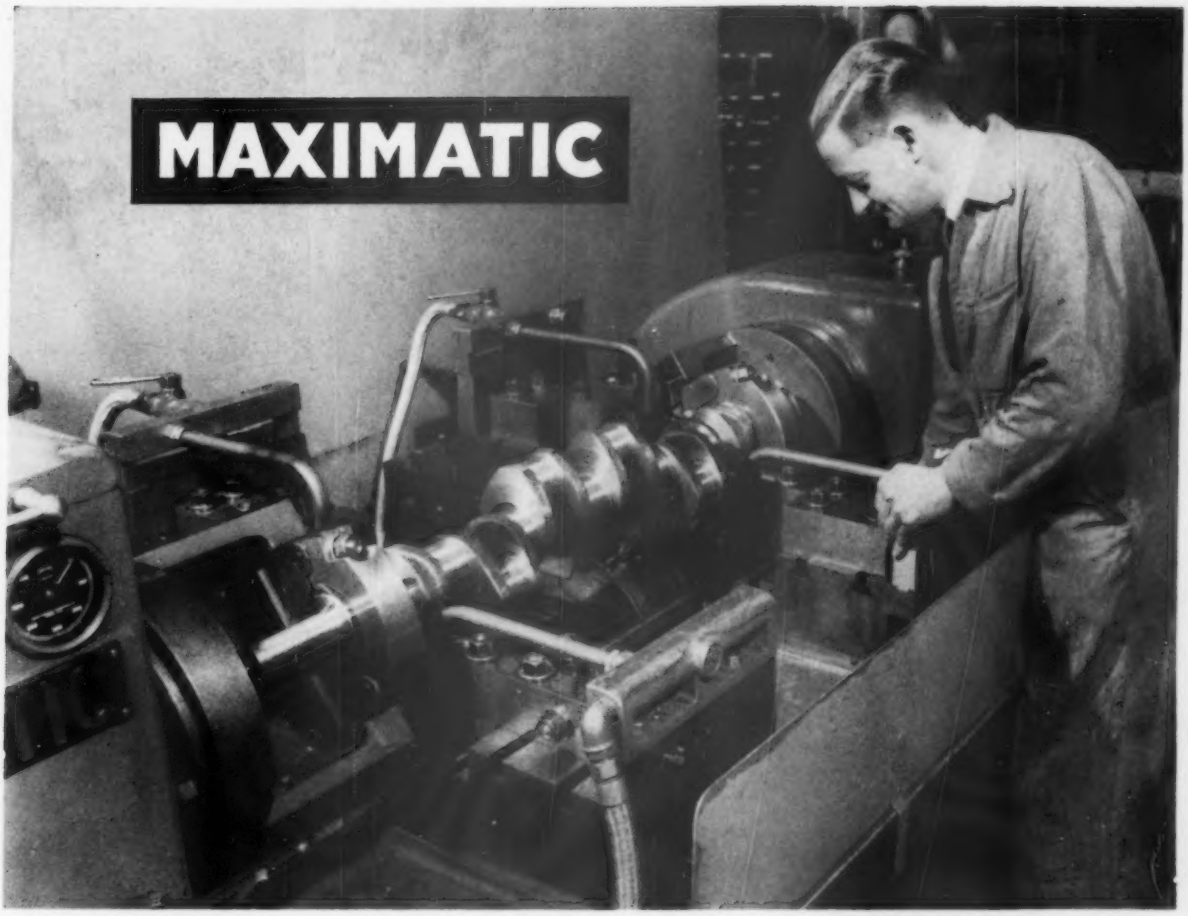

Multi-Tool Lathes for Crankshaft Turning

These automatic multi-tool lathes are widely used in the Motor Industry for turning operations on automobile crankshafts. The illustrations demonstrate the range covered by installations of "Maximatics". The photo above shows machining of A.E.C. Diesel engine cranks and in the lower view turning operations are in progress on Austin Seven crankshafts.

Drummond "Maximatic" lathes operate on a completely automatic cycle after starting the machine. High-output, accurate production is assured and they are equally applicable to the machining of many other components.

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AEC Bridgemaster King-pin assembly fitted with DU bushes.

DU dry bearings are proving their worth on an increasing number of vehicles, including the A.E.C. Bridgemaster. Many leading firms use them in brake and clutch mechanisms, power steering mechanisms, throttle linkages and centrifugal clutch flywheels. They need no lubrication, have a high load-carrying capacity with a lower friction coefficient, eliminate stick-slip, afford protection from abrasive particles, and reduce wear, fire risk and cost. DU dry bearings are made exclusively by GLACIER.

* Glacier DU dry bearings are composed of thin steel strip with a porous bronze coating impregnated with a mixture of a fluoro-carbon plastic (P.T.F.E.) and lead.

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DU Thrust washers fitted as standard equipment on the ¼ ton range of vehicles.

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For details of the range and properties of Glacier dry bearings, write for a free copy of the Designer's Handbook No. 2 to :—

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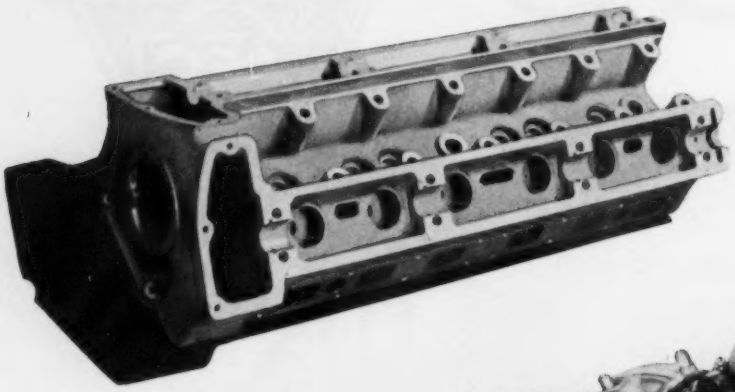

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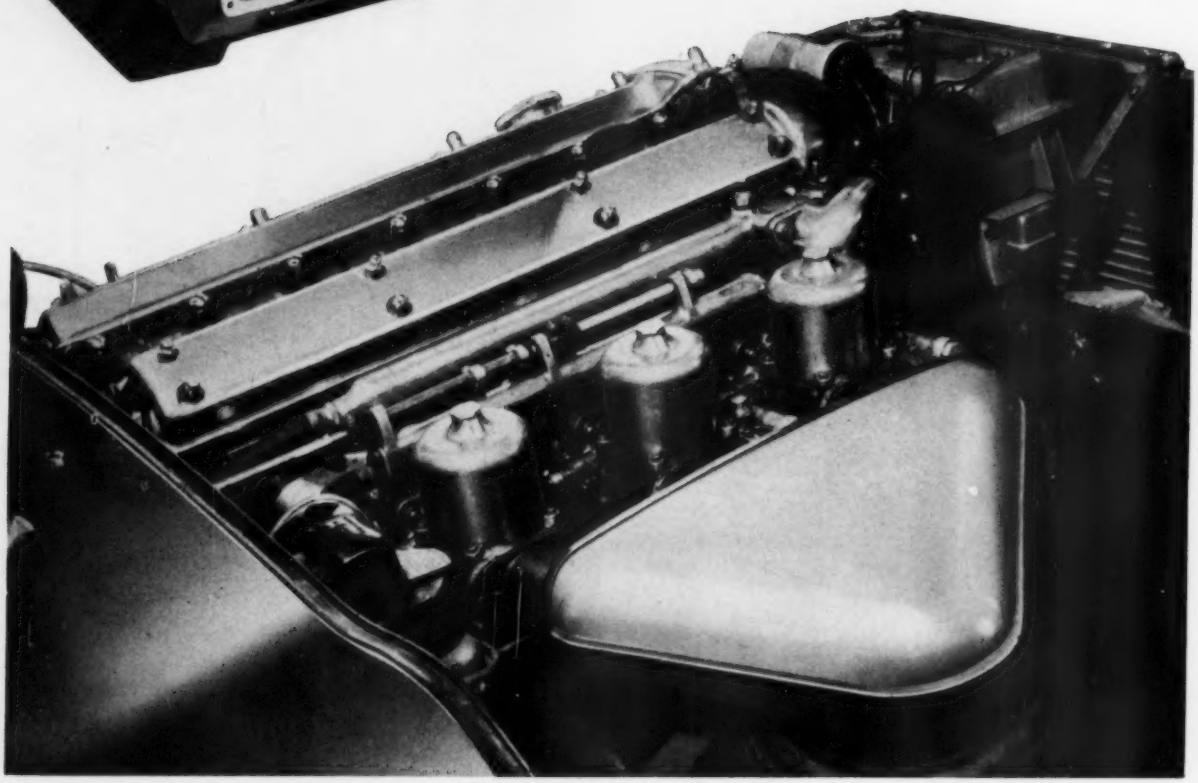

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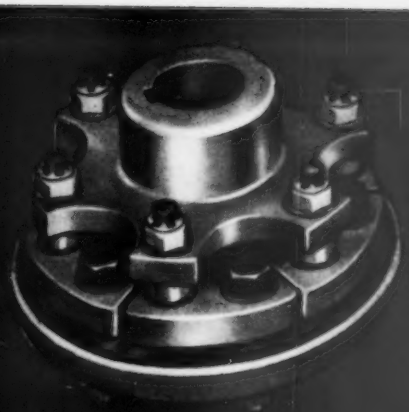

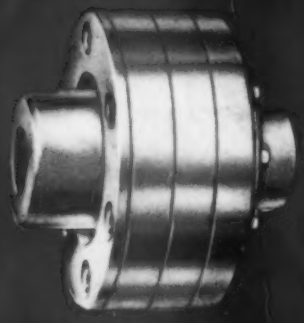

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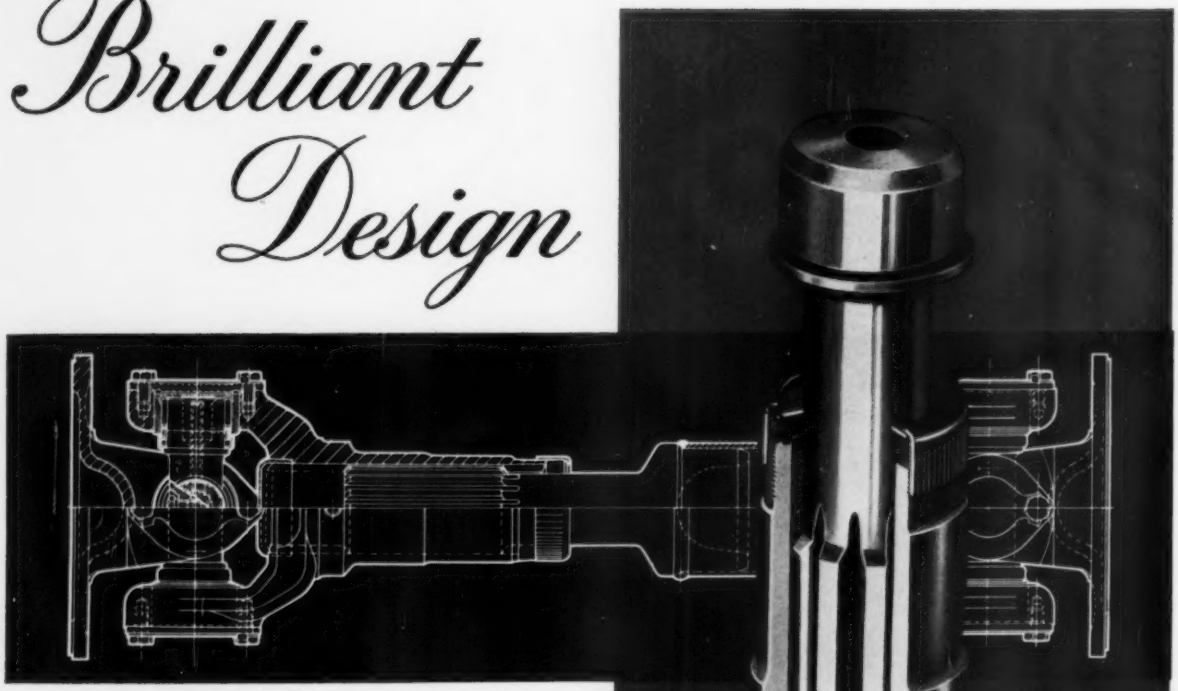

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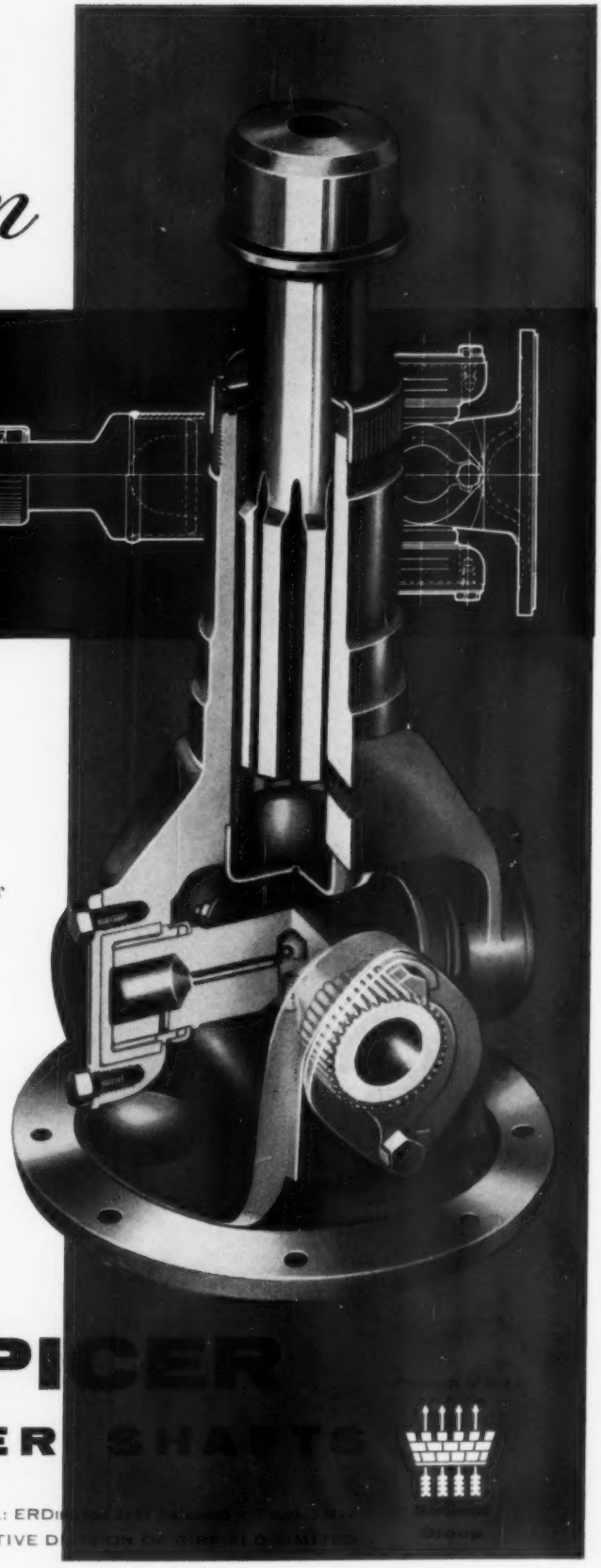

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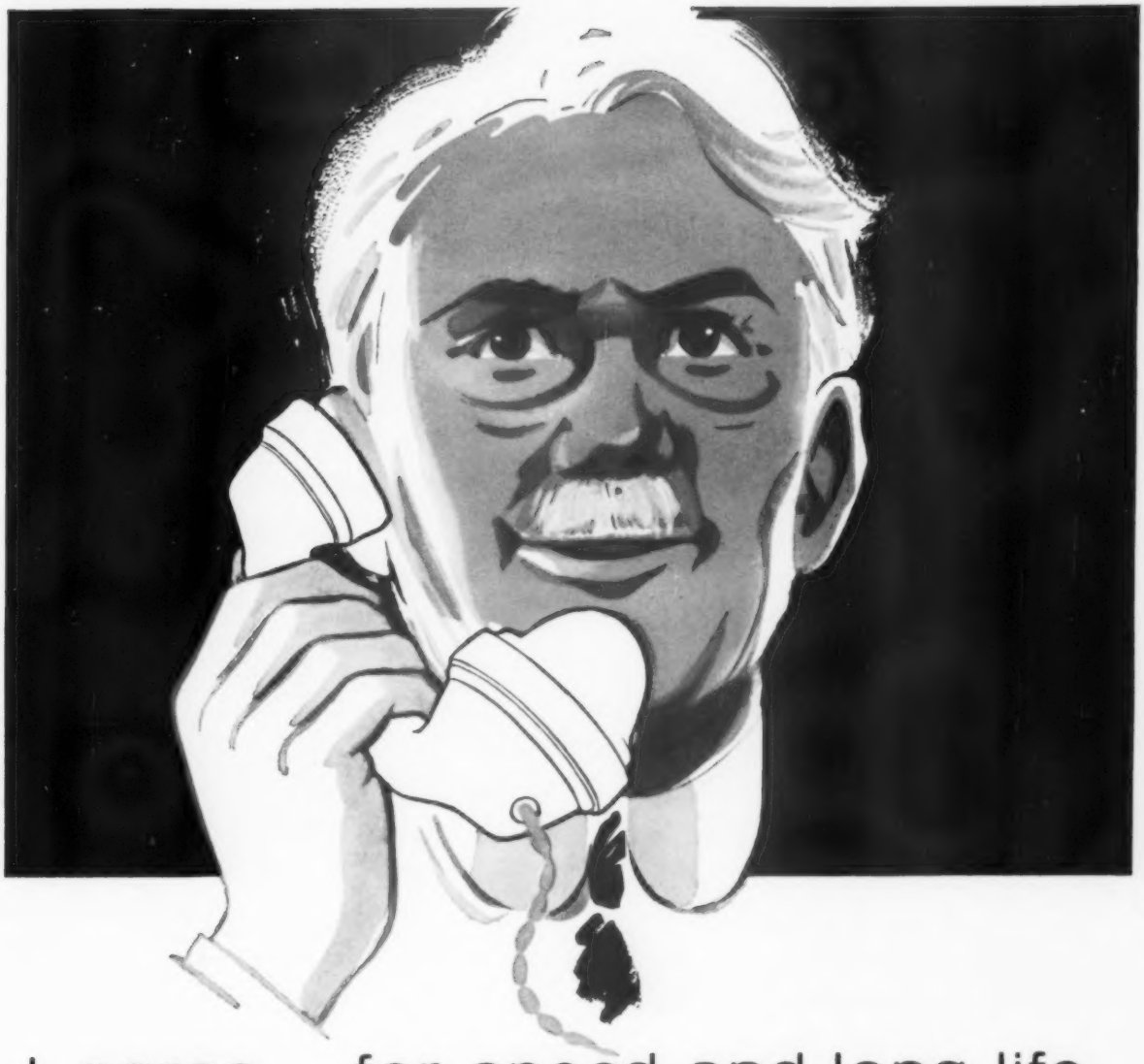

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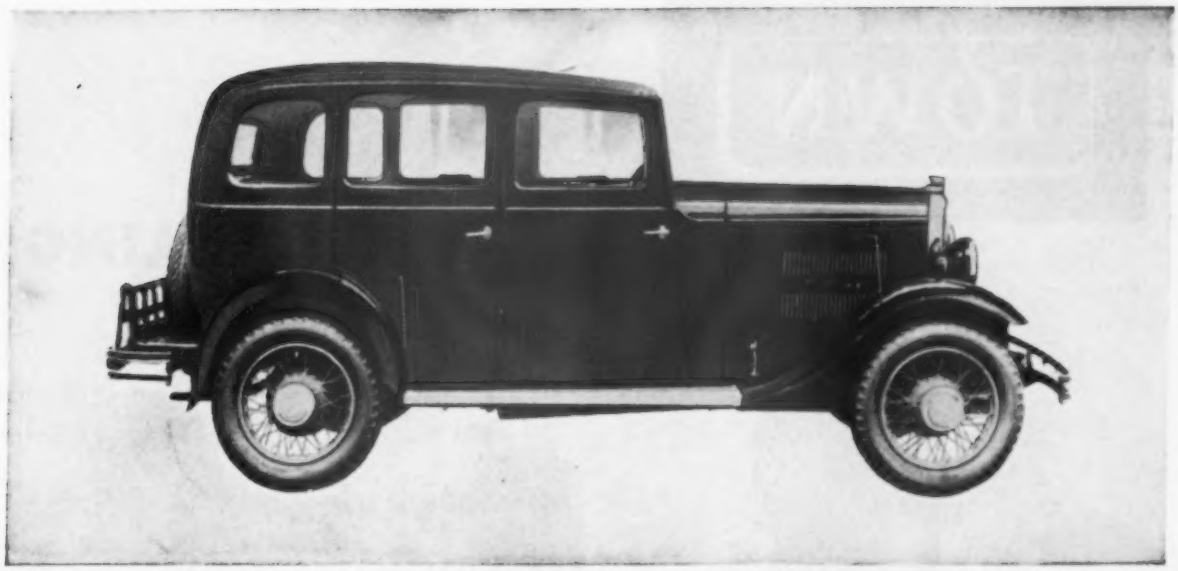

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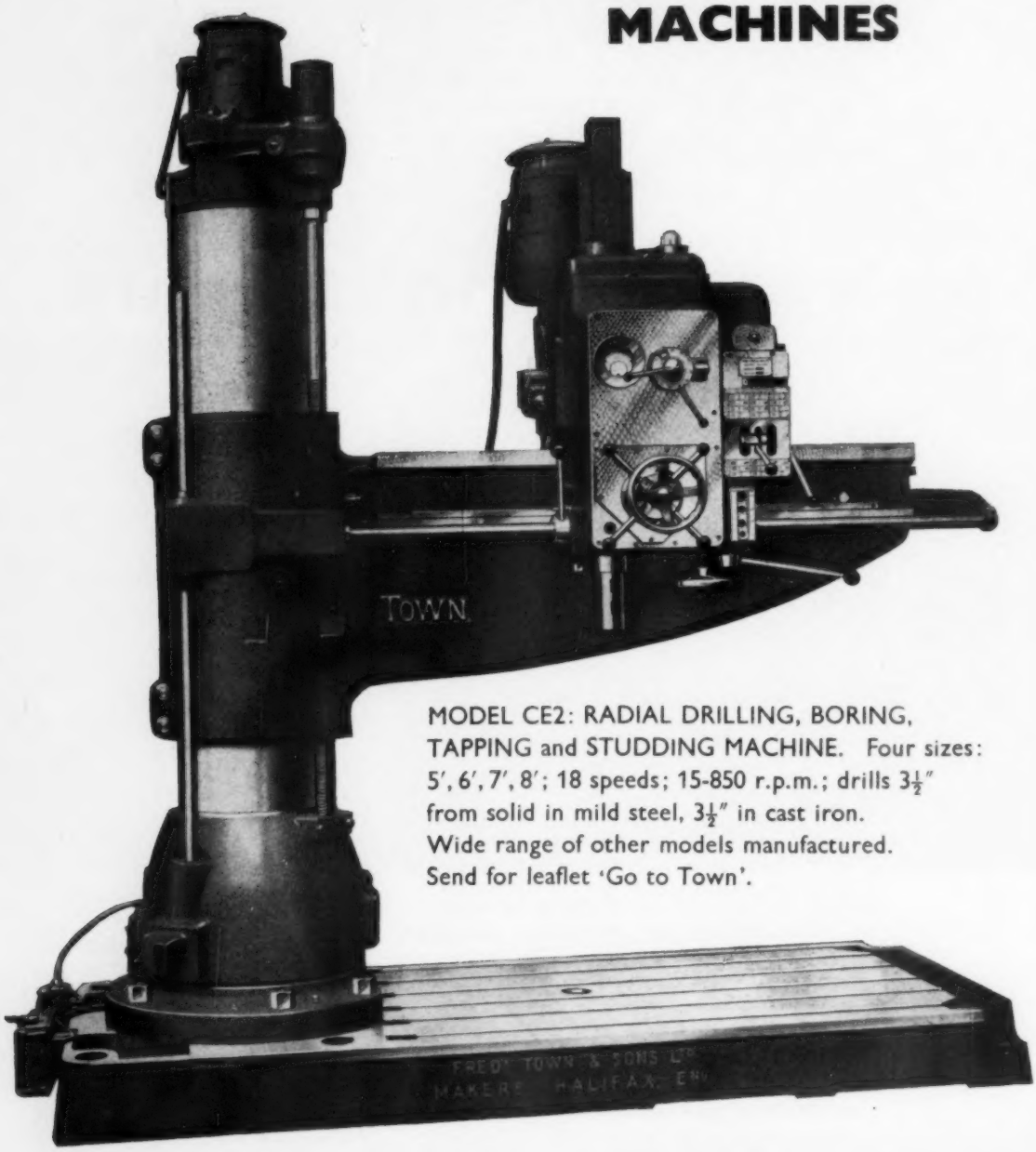

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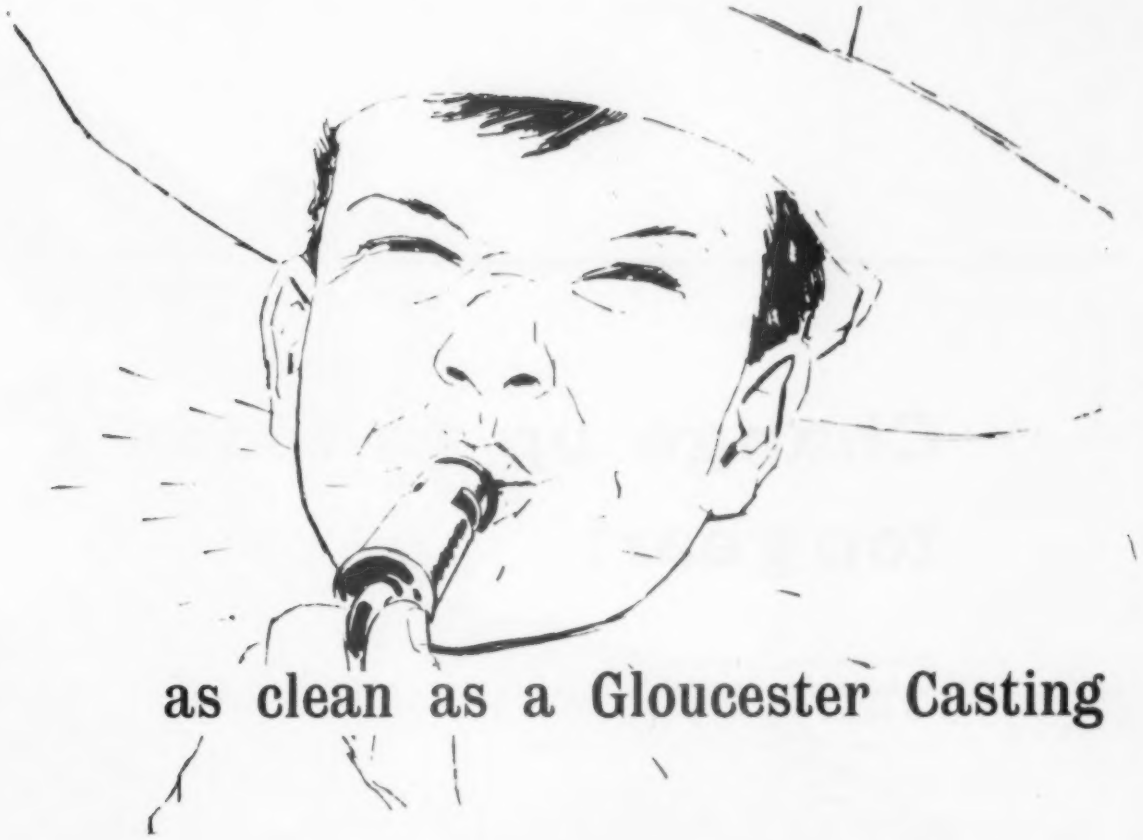

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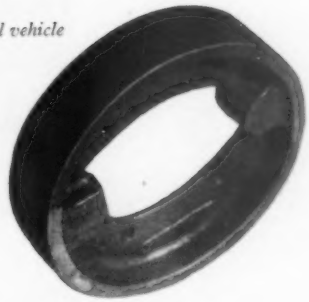

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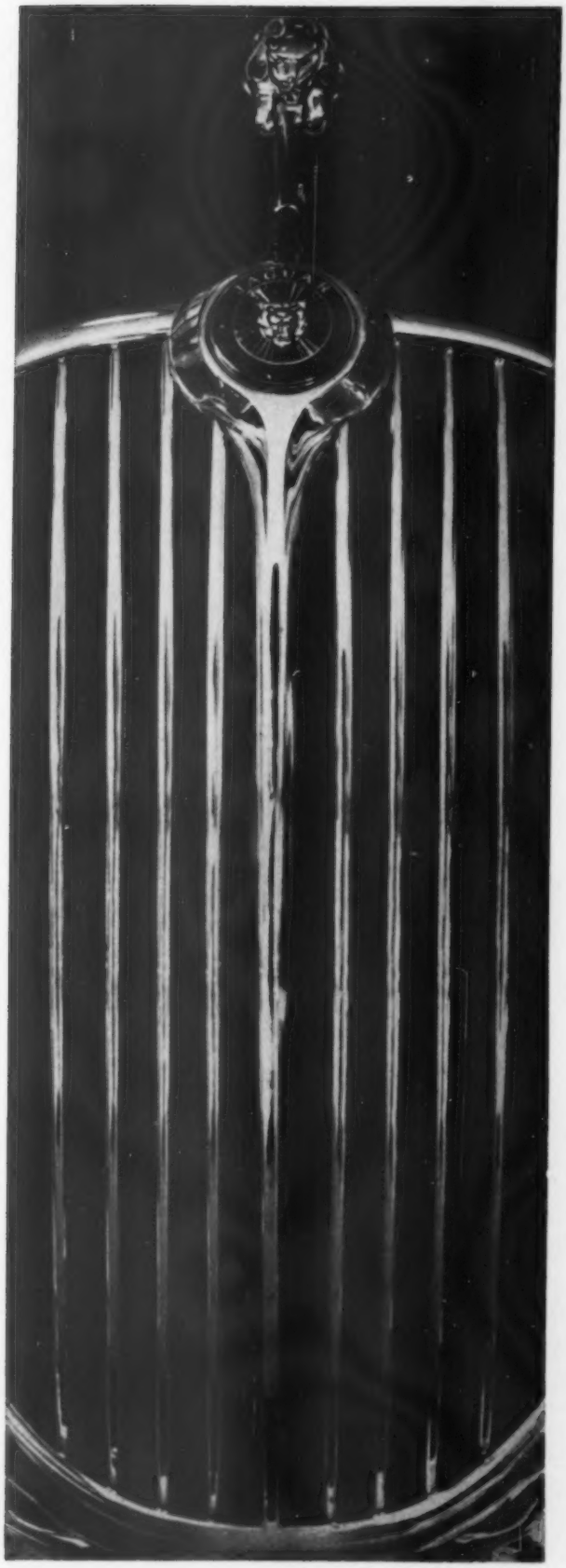

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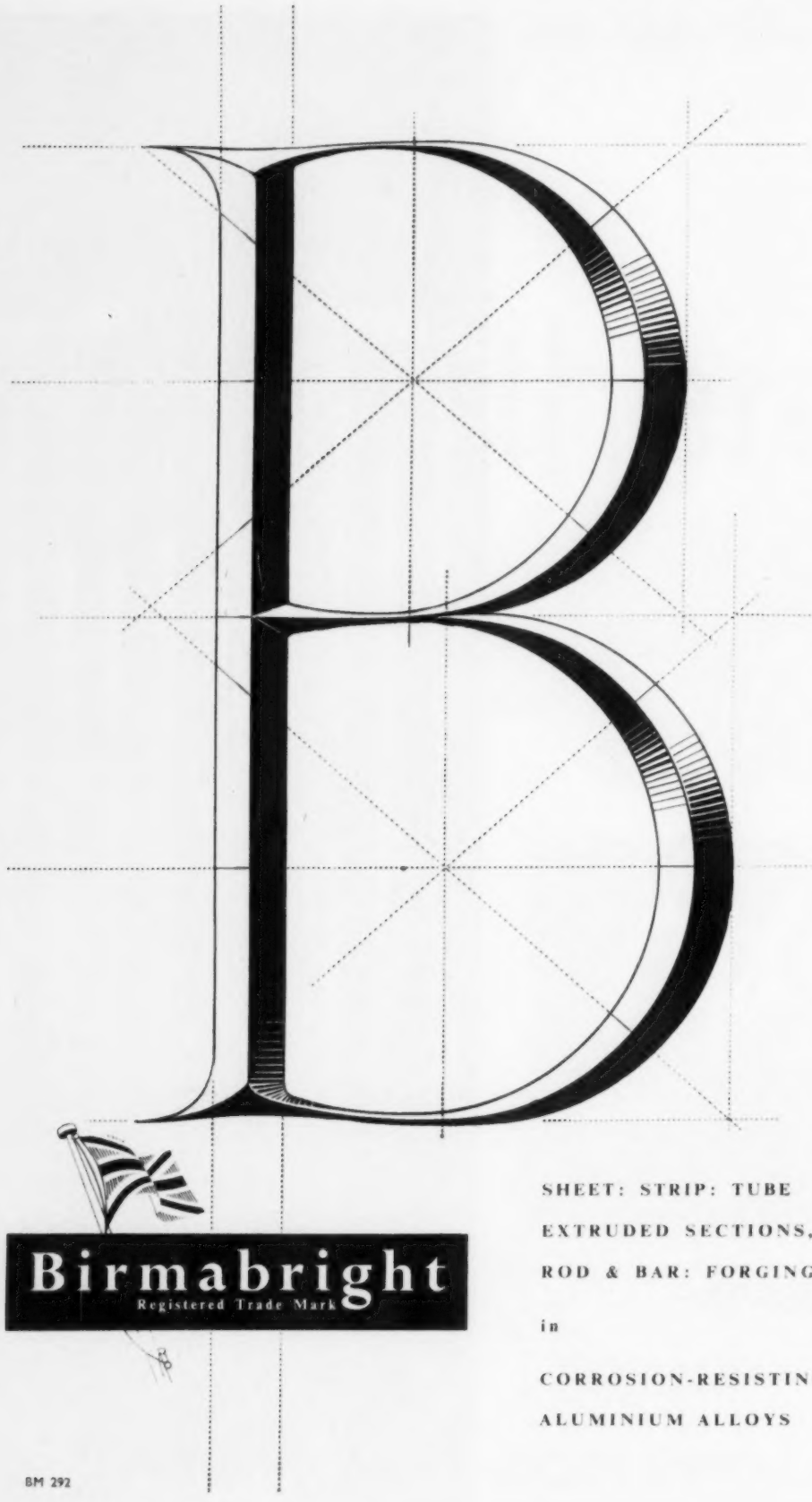

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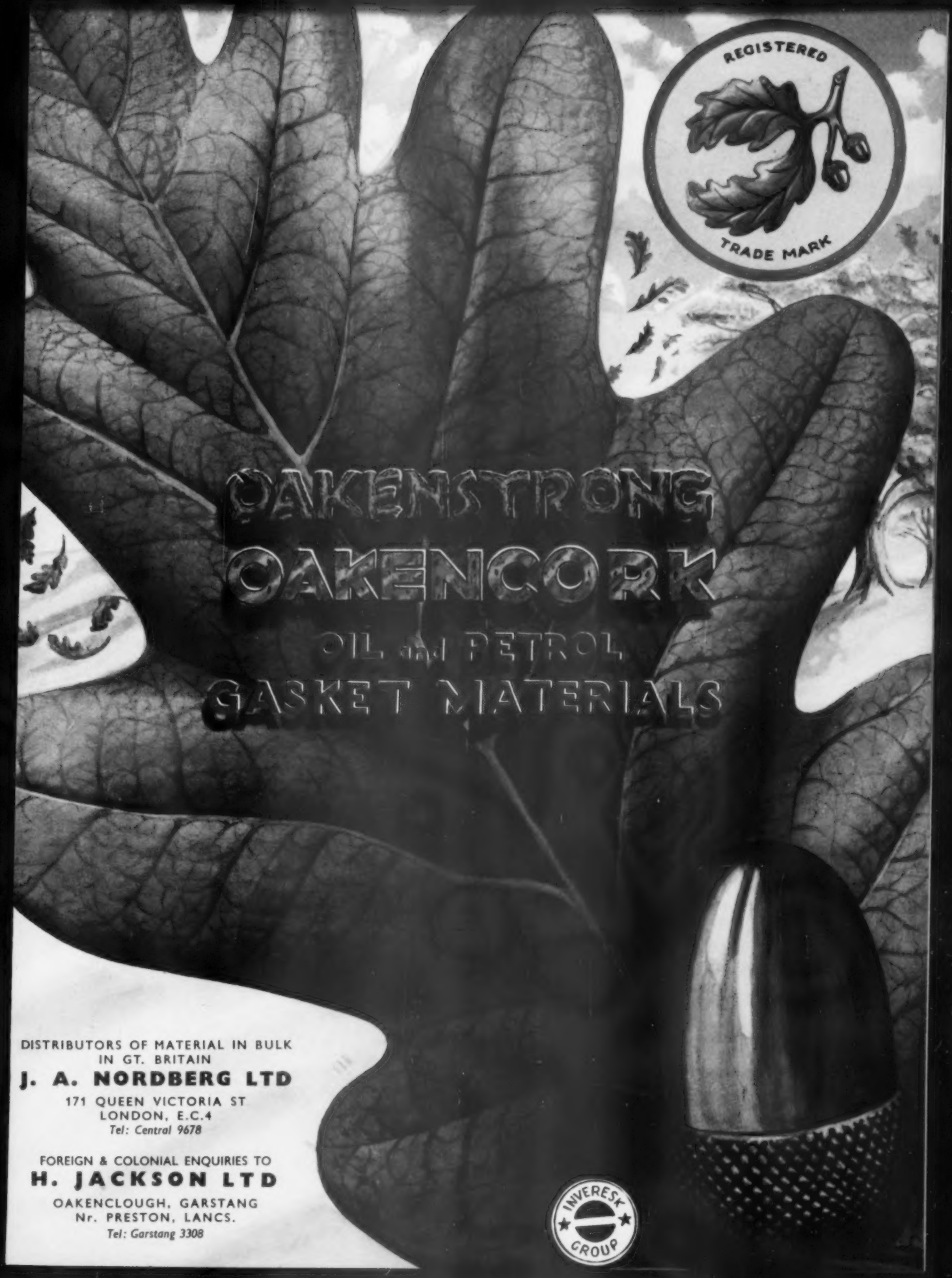

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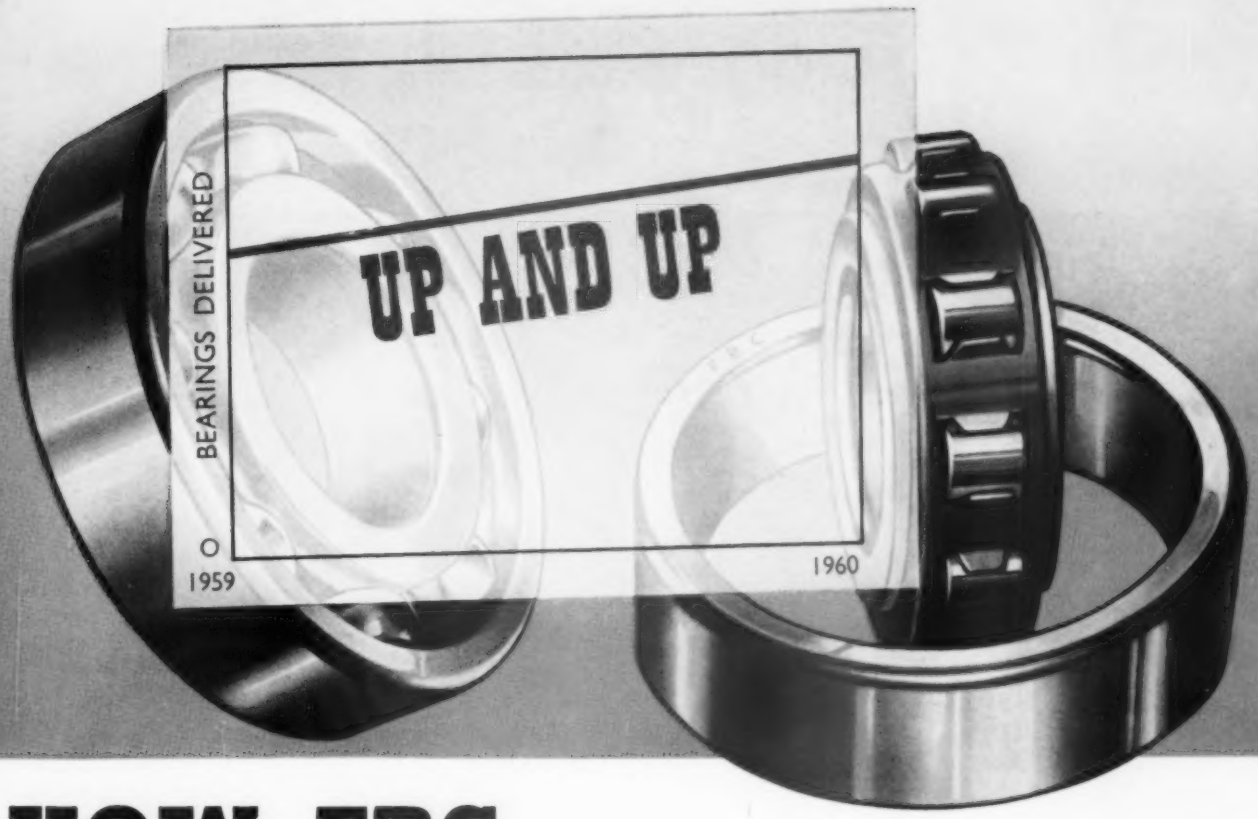

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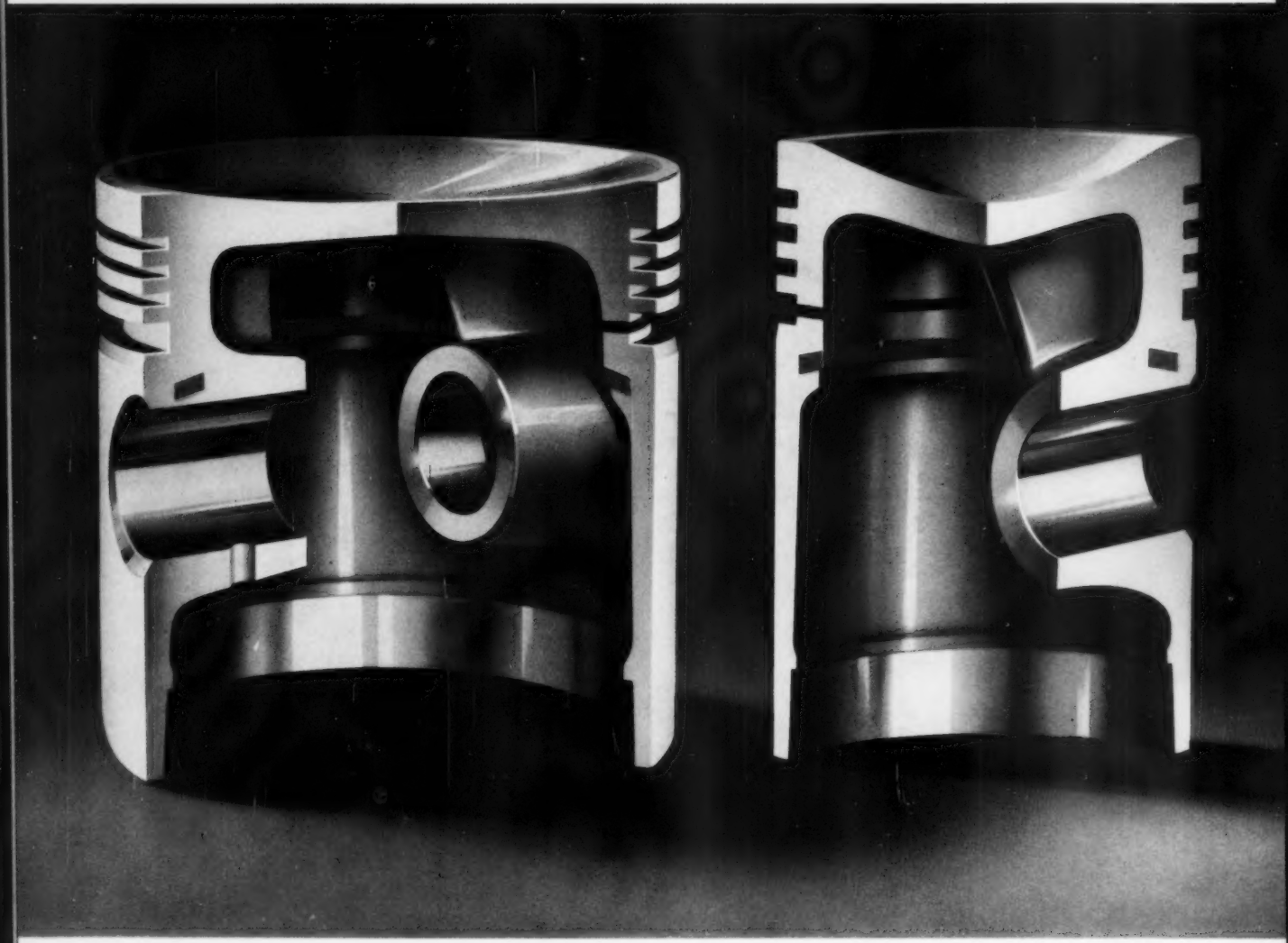

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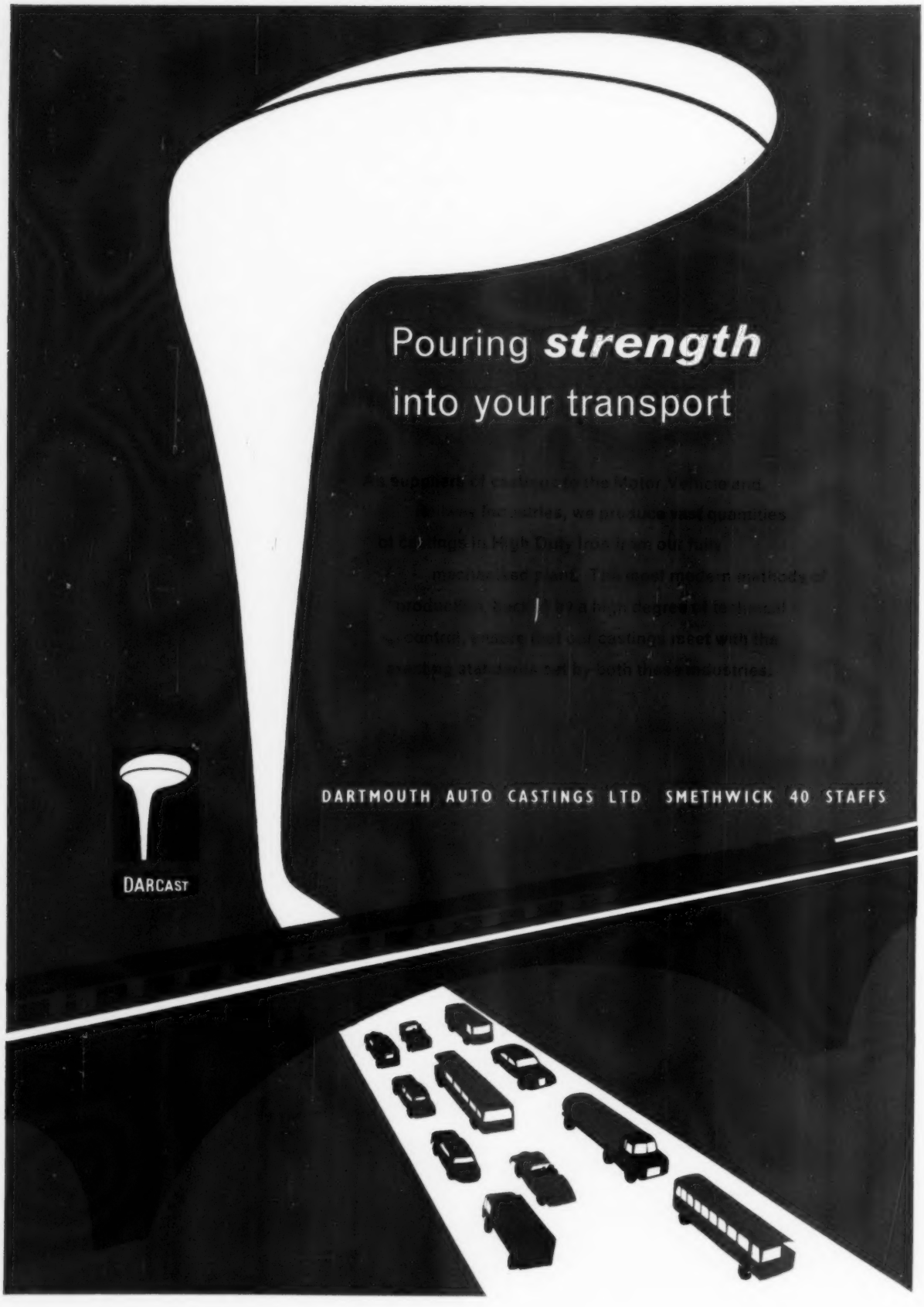
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AUTOMOBILE ENGINEER

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194 Preforming Blanks for Cold Forging Operations *A conveyorized machine line for the continuous automatic production of uniform blanks from bar stock*

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AUTOMOBILE ENGINEER

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Diesel Exhaust Smoke

IN view of the recent extensive publicity on the subject of the adverse, including toxic, effects of exhaust gases, and also the fact that legislation concerning vehicle exhausts has been introduced in some parts of the world, it is obvious that other governments may soon be forced into taking similar legislative action. In the vehicle manufacturing industry it is well known that, in the open, the diesel engine exhaust is virtually harmless, even if judged by the standard of that of petrol engines, and therefore manufacturers of diesel engines tend to dismiss the problem as of no concern to them. However, there are no grounds for this complacency: the essential fact is that the public—who in general are not accurately informed on this subject—cannot fail to observe clearly the black smoke that sometimes issues from diesel engines, whereas the more noxious effluent from petrol engines is, of course, invisible. It follows that public outcry tends to be directed against the diesel engine, and therefore effective action must be taken to overcome the smoke problem, despite its irrelevance to, for example, lung cancer.

Those who have made a thorough study of the subject—and, as all know, this includes not only the engine and injection equipment manufacturers but also the oil companies—have found that the really severe cases of smoking exhausts are almost always attributable to either faulty maintenance or interference by the drivers with the fuel delivery controls. Since it is only these very bad cases of smoke emission that give rise to public outcry, manufacturers tend to take the view that this is solely the concern of the maintenance engineers or drivers. In a sense this attitude is justified but, in that failure to eradicate the nuisance could lead to legislative action having significant repercussions on design and manufacture, those who make engines would be well advised to take steps to overcome the problem.

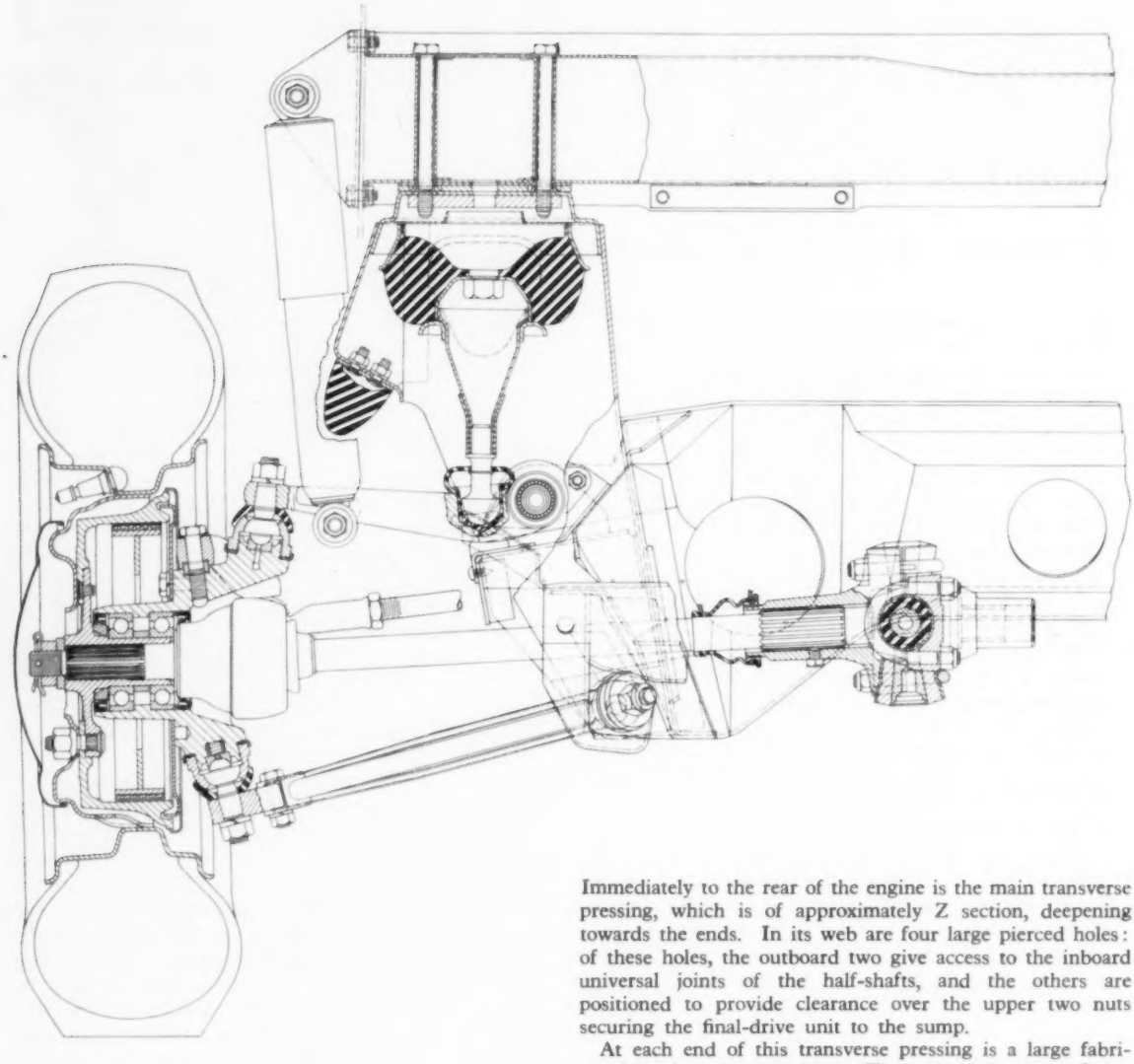
An indirect remedial measure that would help is the wider education of operators regarding the importance of systematic maintenance. For example, arrangements might be made for this aspect to be given more prominence in courses already available at local technical colleges. Manufacturers, in their literature, might place more emphasis on the advantages to operators of good maintenance of injection equipment and also the penalties of failure in this respect. So far as direct action is concerned, results can be obtained only by relatively long term research. It seems that the most promising field is that of combustion: in particular, a most careful study is needed of the later stages of the combustion process, in which there is less air available and therefore more carbon is likely to be formed.

Perhaps the most common cause of smoke is overfuelling. For the circumvention of difficulties due to human shortcomings, injection equipment is now widely available in which it is very difficult—indeed virtually impossible—for the driver to tamper with the maximum delivery stop. Obviously, the control for the excess fuel device, for starting, should not be in the cab. As regards maintenance, among the defects to be avoided are leaking valve seats in the injection equipment, and sticking needles and blocked holes in the nozzles. In the fuel pumps, sticking delivery valves may cause overfuelling, as also can wear of the delivery valve unloading collar. Filter maintenance, too, is significant, and the importance of using the filter elements recommended by the engine manufacturers is sometimes overlooked. Inadequate compression in the engine cylinder can cause smoking: this defect, of course, may be due to leaking valves, sticking piston rings, or choked air intake filters. Smoking after overhaul is most likely to be caused by incorrect timing or wrong assembly of injection equipment. Nor should it be overlooked that faulty timing is sometimes attributable to wear of a coupling.

These are only some of the defects that can cause smoking exhausts. To list them all, together with their remedies, would be far beyond the scope of this article. Our aim is simply at showing that there is a pressing need for positive action to rectify the situation, and that this action should be taken jointly by the oil companies and operators after frank, but confidential, discussions on this subject. Since both the manufacturers and the oil companies are loath to take the initiative for fear of offending their customers, perhaps in the case of Great Britain the approach to the operators could be made by the S.M.M. and T. who, as is well-known in the industry, already have done some good work in this field.

B.M.C. ADO 15

Part II: The Front and Rear Sub-Frames and Suspension Systems, Steering Gear and Brakes



IN the introductory section of Part I of this article, page 125, reference was made to the fact that there is a sub-frame at each end of the body unit of the ADO 15 vehicles. Both sub-frames perform the function of reinforcing the structure, but whereas that at the front carries the engine-transmission unit and the suspension, its counterpart at the rear merely provides suspension anchorage points. In consequence of these different duties, the front sub-frame is the more complex of the two; both are fabricated from steel pressings.

The pressings of the frontal unit are mainly of 13 s.w.g. material, reinforced locally in highly stressed regions. Immediately to the rear of the engine is the main transverse pressing, which is of approximately Z section, deepening towards the ends. In its web are four large pierced holes: of these holes, the outboard two give access to the inboard universal joints of the half-shafts, and the others are positioned to provide clearance over the upper two nuts securing the final-drive unit to the sump.

At each end of this transverse pressing is a large fabricated bracket of turret shape. The ends of the upper flange of the main pressing are fillet welded to the brackets, whereas the web is extended and is lap welded to their rear faces. These turret brackets form the main pick-up points for the body shell, and also house the spring units and carry the pivot bolts of the front suspension links. The section of each is an outward facing channel, the upper 4 in and top of which are closed by additional pressings. Reinforcing plates are welded within the channels, to provide increased support for the pivot pins of the links. In the web of each turret bracket, near the bottom, is a lipped hole, through which passes the half-shaft; the hole is of slotted form to provide working clearance when the shaft articulates under suspension movement, and to permit the shaft to be readily installed after the power unit is mounted on the sub-frame. Other pierced holes, in the flanges of each

bracket, towards the top, facilitate installation of the spring.

Extending forward from the lower end of each turret bracket is a short side member. These members are of a tilted W section that tapers in width and depth towards the front: the outboard and uppermost flange of the W is vertical, the web adjoining it is horizontal, and the second web is inclined inward at 30 deg to the vertical; the inboard flange is horizontal. At the front, the two horizontal faces of each side member are welded to the upper and lower flanges of a transverse member of channel section. Brackets at the ends of this member form anchorages for the radius rods of the front suspension. Towards each end of the cross member is a hole for the attachment of the body.

Two pressings are spot welded, one at each side, to the lower flange of the main transverse pressing of the sub-frame. They extend horizontally rearward, are of triangular shape, as viewed in plan, and have a cross-section approximating to a flattened and inclined Z. Their edges are lipped, to increase the stiffness. In the middle of each pressing is a large lightening hole, which also has a lipped edge, and at the rear end of each are two small holes, whereby it is bolted to the undersides of the floor and the toeboard of the body.

Spot welded to the front of the dash structure of the body shell is a transverse member, of top hat section, reinforced at the ends to form seatings for the tops of the turret brackets. A steel packing plate is interposed between each bracket and this member. Two pairs of long bolts pass vertically through spacer tubes welded within the member; the bolts of each pair screw into a captive anchor block under the top pressing of the appropriate bracket.

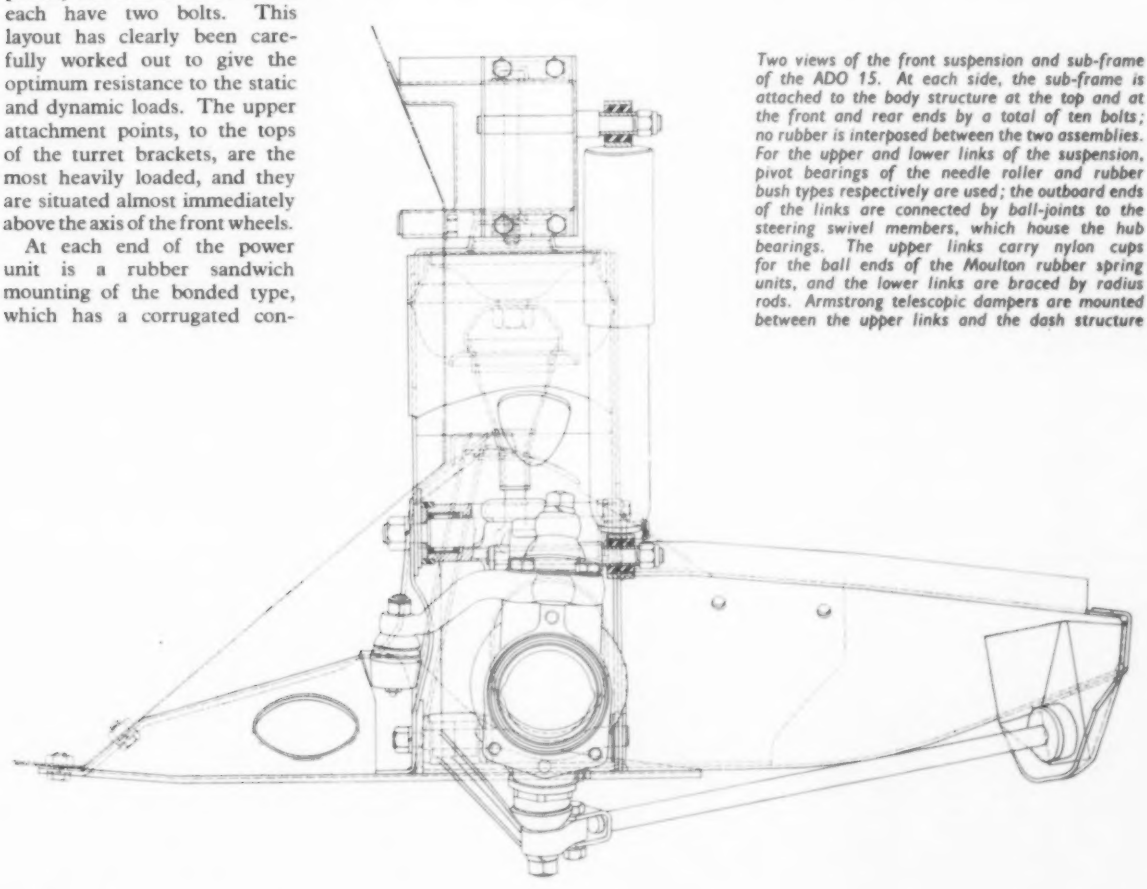
The body is thus rigidly attached to the sub-frame at six points, the rear four of which each have two bolts. This layout has clearly been carefully worked out to give the optimum resistance to the static and dynamic loads. The upper attachment points, to the tops of the turret brackets, are the most heavily loaded, and they are situated almost immediately above the axis of the front wheels.

At each end of the power unit is a rubber sandwich mounting of the bonded type, which has a corrugated configuration to provide adequate control of engine movement. Since the mountings are bolted to the inclined web of the sub-frame side members, the rubber acts in combined compression and shear. The method of attaching the mountings to the engine was described in Part I, but mention should now be made of the stabilizing arrangements. Rocking under torque reaction is restricted mainly by a rod, with rubber in its end-fittings, installed horizontally between the top of the cylinder block and the dash structure. However, the exhaust pipe, which is clamped rigidly to the manifold and to the final-drive casing, forms a secondary restraining link: further aft, at the point where it enters the silencer, it is resiliently attached to the underside of the body floor.

Front suspension

In its general design, the front suspension clearly stems from the system that has proved so satisfactory on all models of the Morris Minor. Basically, the layout is of the double transverse link type, in which the longitudinal dynamic loads are reacted at each side by a semi-trailing radius rod, which has the effect of converting the lower link into a wide-base wishbone.

A noteworthy feature of this system is that a limited degree of longitudinal resilience is provided, by the use of rubber in the anchorages of the radius rods and in the pivot bearings of the lower links. At a time when the American motor industry is regarded by many as exemplifying the progressive outlook, it is gratifying to realize that the advantages of such resilience, now being publicized by more than one transatlantic firm, were appreciated by the Nuffield organization as much as 15 years ago, when the development of the Morris Minor was being undertaken.



Two views of the front suspension and sub-frame of the ADO 15. At each side, the sub-frame is attached to the body structure at the top and at the front and rear ends by a total of ten bolts; no rubber is interposed between the two assemblies. For the upper and lower links of the suspension, pivot bearings of the needle roller and rubber bush types respectively are used; the outboard ends of the links are connected by ball-joints to the steering swivel members, which house the hub bearings. The upper links carry nylon cups for the ball ends of the Moulton rubber spring units, and the lower links are braced by radius rods. Armstrong telescopic dampers are mounted between the upper links and the dash structure

In spite of the basic similarity of design, there are several major differences between the ADO 15 suspension layout and that of the earlier car. First, variable-rate rubber springs are employed instead of torsion bars which, of course, provide a virtually constant rate at the wheel over their operating range. Secondly, it will be recalled that, on the Morris Minor, the upper transverse links serve also as the arms of lever type hydraulic dampers. In contrast, the dampers of the ADO 15 are of the telescopic type, and the upper links incorporate the lower abutments of the springs.

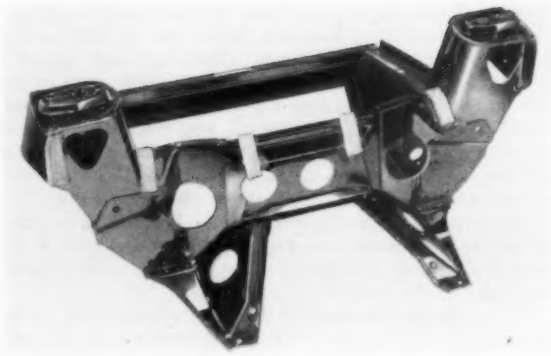
Because the change from torsion bar to rubber springs is a significant one, the reasons for it are of especial interest. Of these reasons, the most important is the well-known fact that the smaller and lighter the car, the greater is the ratio of laden to unladen weight; the higher this ratio, of course, the more difficult it is to select a single-rate spring that is neither too soft when the vehicle is fully laden, nor excessively firm in its action in the lightly loaded condition. In the case of the ADO 15, which has a kerb weight of only 1,260 lb, a variable-rate spring was, therefore, regarded as a virtual necessity. However, a coil spring of this type would have had too great an axial length for convenience of installation, while a compound torsion bar spring with more than one rate was not practicable.

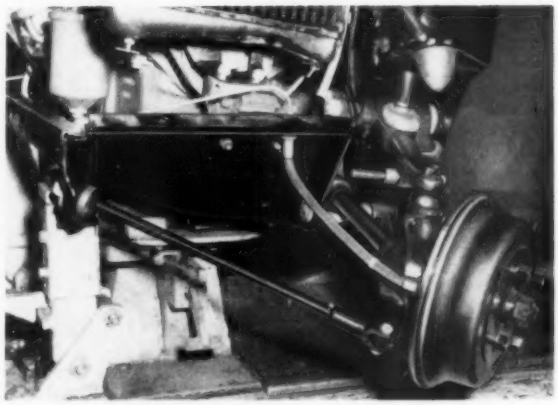
Rubber appeared to be the most promising alternative springing medium, so Moulton Developments Ltd.—an associate company of B.M.C.—was requested to design and develop suitable units. From the outset, it was clear that a light, cheap and compact suspension unit could only be evolved if its total working movement was limited to about 1 in; otherwise, the amount of rubber necessary would be uneconomic. It was, therefore, necessary to employ a lever system, whereby an adequately long wheel movement could be obtained with a small deflection of the spring. An additional advantage of rubber springs was that the relatively high hysteresis of the material would permit the use of lower-duty dampers than would be satisfactory with steel springs. It is appropriate to mention here that the Moulton springs, which will be described later, have proved entirely satisfactory in service; the production units are made for B.M.C. by the Dunlop Rubber Co. Ltd.

The two pairs of suspension links are situated approximately transversely — they actually have a slight leading angle to bring the inboard ends of the lower links clear of the half-shafts on full rebound — with an appreciable divergence of their outer ends, as viewed from ahead. They are ball jointed to the steering swivel members, which house the wheel hub bearings; as previously stated, their inboard ends pivot on spindles carried in the turret brackets of the sub-frame. As full bump is approached, the upper links come into contact with paraboloidal rubber buffers mounted on the turret brackets. The purpose of these buffers is to limit the maximum stress in the rubber springs. Rebound of the wheels is checked not only by stops within the dampers but also by contact between the upper links and additional rubber buffers carried on the turret brackets.

Each upper link is a forging of 55 ton steel and its length, between the pivot axis and the centre of the ball joint, is 7 in. As viewed in plan, it is of T shape; the bar of the T is at the inboard end and is in the form of a long boss that houses the pivot bearings. Because of the high loadings imposed by the spring units, these bearings are of the Torrington needle roller type; they are installed in counterbores, and the nominal length and diameter of the rollers are $\frac{1}{2}$ in and $\frac{3}{32}$ in respectively.

The rollers run on a hardened steel pin of $\frac{3}{4}$ in overall diameter, stepped down to $\frac{1}{2}$ in diameter at the ends, which pass through the webs of the turret bracket of the sub-frame. Between each web and the adjacent shoulder of the pin is a hardened steel thrust washer, which is trapped in position when the nuts of the pin are tightened on assembly. Both bearings are lubricated through a single grease nipple in the pivot boss. To retain the grease, a synthetic rubber sleeve is fitted over a smaller-diameter portion at each end of the boss; in its free state, the sleeve projects beyond the thrust washer, and so is lightly compressed against the web of the turret bracket when the securing nuts are tightened.





Top and middle: Views from the rear and front of the front sub-frame. The turret brackets, the front cross member and the pressings that project to the rear are bolted to the body shell. Bottom: As on the Morris Minor, rubber is embodied in the leading anchorages of the radius rods of the front suspension lower links; the resulting longitudinal resilience minimises the transmission of road shocks to the body shell

Since the leverage ratio adopted for the spring units is 5:1, the distance from the pivot axis of the link to that of the spring seating is 1·4 in. This seating is a nylon cup, pressed into a blind hole drilled downward into the link. The axis of this hole is normal to the plane through the pivot axis, which is horizontal and longitudinal, and the centre of the ball joint at the outer end of the link.

A 32 ton steel is employed for the lower links, which are I-section forgings with an effective length of 10·1 in. At its inboard end, each link pivots on a bearing of the shouldered rubber type, mounted on a $\frac{1}{2}$ in diameter spindle. The spindle is carried in the webs of the turret bracket and has a dip near the middle of its length, ahead of the bearing, for working clearance under the half-shaft. Near the outboard end of the link, on the leading face, is a lug for the pivot bolt of a clevis welded to the trailing end of the steel radius rod, the diameter of which is $\frac{1}{2}$ in.

Whereas the clevis attachments do not embody any rubber, the leading end of each rod is connected to its bracket at the front of the sub-frame by an orthodox rubber sandwich type fitting. The radius rods have an effective length of $16\frac{1}{8}$ in, measured between the geometric centres of the fittings and those of the lower ball-joints, and they are inclined laterally at an angle of 35 deg to the longitudinal axis of the car. The base of the wishbone formed by the link and the rod is longitudinal and horizontal, and has a length of $16\frac{1}{4}$ in.

Both the upper and lower ball-joints are of the type in which two different spherical radii are employed. These joints are easier to produce than are the single-radius components and are more economical in respect of material. In each case, the portion of the ball adjacent to the pin has the larger radius. Each pin is taper mounted, in the conventional manner, in the end of its link, and each ball seating is in two parts. The housing for the larger-diameter portion of the ball is of steel, machined from the solid, and is screwed on to the swivel member, and secured by a lock-washer. To obtain the correct adjustment of the joint, shims are fitted between the member and the housing.

The seating for the smaller-diameter portion of the ball is a hard steel pad, which is partially recessed into the swivel member. Both the pads are a sliding fit in their holes, and that of the lower joint is held in contact with the ball by a helical spring of square-section wire, which applies a thrust of 60 to 65 lb. This spring loading obviates any risk of rattle under the reversals of loading to which the joint is subjected when the suspension operates. The ball-joints are lubricated by means of grease nipples on the swivel members, and the grease is retained by a moulded rubber cup that fits closely round each pin and the outside of the screwed-on ball housing.

Each swivel member is a malleable casting, and its form is approximately a hollow truncated cone, with its axis horizontal; at the top and bottom are projecting bosses for the ball-joint housings. The skirt of the casting prevents road dirt from being flung directly at the inboard seal of the hub bearings. On top of the member, outboard of the upper ball-joint, is a flat face to which the steering arm—a 55 ton steel stamping—is attached by two $\frac{1}{8}$ in diameter bolts secured by a double tab washer. For accurate location of the arm, the bolt holes in both components are counterbored to receive sleeve dowels, pressed into the member.

On this member, immediately outboard of the steering arm, are machined a shallow male spigot and a shoulder. The back-plate of the brake assembly, which is located by the spigot and by a dowel, is secured against the shoulder by three $\frac{5}{16}$ in diameter set-screws, fitted with spring washers. In the outboard portion of the casting are housed two Ransome and Marles ball bearings, the outer races of which are pressed in, one from each end, against a circumferential rib that forms a double shoulder. These are the hub bearings, and they are of the angular-contact type, mounted in opposition to resist axial thrust in both directions. Their overall diameter is 62 mm, bore $1\frac{1}{4}$ in and width 16 mm. Because each outer race can only react thrust towards the shoulder, no means other than the interference fit is necessary to secure them.

The hub member, an En.17W steel forging with an external flange—for the wheel fixing studs—and a splined bore, is pressed from the outboard side into the inner races of the bearings, which are separated by a spacer ring. Between the inner race of the outboard bearing and the flange of the hub member is a second spacer ring. The splines on the stub axle, which is integral with the outboard portion of the universal joint, are push fit in those in the bore of the hub forging; the inner race of the inboard bearing projects slightly beyond the hub member, and registers against a shoulder on the joint. A thick washer and a castellated nut with split-pin are assembled on to the end of the stub axle, to clamp the hub bearing assembly against this shoulder. On each side of the bearing assembly is a lip type oil seal; both seals are housed in the swivel member and, whereas the inboard one bears on the driven member of the universal joint, the outboard one bears on the second spacer ring just mentioned.

Measured from the horizontal, the lateral inclination of the king-pin axis—that is, the axis through the geometric centres of the two ball-joints—is $81\frac{1}{2}$ deg. This axis meets the road at a distance of $1\frac{1}{8}$ in inboard of the major axis of the tyre contact patch, and the castor angle is 3 deg positive. An accompanying reproduction of the general arrangement drawing of the suspension clearly illustrates the compactness of the layout of the swivel and hub assembly.

The rubber spring units are installed with their axes vertical. Each consists basically of an inner and an outer part-conical member, both of which are of steel, to which is bonded an annular rubber element; the cones and the varying thickness of the element across the section are the primary means of obtaining the desired change in rate with suspension movement. Both conical members are 12 s.w.g. pressings, and the outer one registers in a seating at the top of the turret bracket of the sub-frame. No other securing device is necessary, because the spring is under compression at all times, even when the rebound stops are in operation.

Into the inner conical member is pressed a strut, in the form of an elongated pressed steel cone, of 16 s.w.g. material. Welded to the strut, adjacent to the rubber element, is a dished annular pressing. This is known as the *co-operating plate* and it assists in obtaining the desired rate characteristics, by enforcing an additional deflection on the rubber as the bump travel increases. The truncated apex of the strut terminates in a short parallel portion into which is pressed the shank of the hardened En.8 ball-end that fits into the nylon seating cup in the upper link of the suspension. Where it embraces the shank of the ball-end, the stem is reinforced externally by another steel pressing. A synthetic rubber gaiter seals the ball-joint: it fits snugly round the shank of the ball-end and has an in-turned lip

A Moulton spring of the type used in the suspension systems of the ADO 15. In these springs, rubber forms the resilient medium; the element is bonded to two pressings, both of part-conical form

that fits under an out-turned lip round the top of the cup. On assembly, the ball-end is coated with a special grease, and no further lubrication is required between overhauls.

The section of the rubber element of the spring at the static load position is illustrated, as also is the external form when the spring is unladen. A total wheel movement of 5⅛ in is provided by the springing, of which 3⅛ in comprises the bump range. One of the accompanying illustrations is the load-deflection graph for a typical Moulton spring. Although this graph gives an accurate indication of the variable-rate characteristics and of the hysteresis effect, the curves for an actual ADO 15 spring differ slightly from those shown. However, the B.M.C. has revealed that, in the static load position, the springs give a wheel rate of about 118 lb/in, and the maximum wheel load—at full bump—is in the region of 1,100 lb. Each spring weighs approximately 2 lb, of which 64 per cent is stated to be rubber.

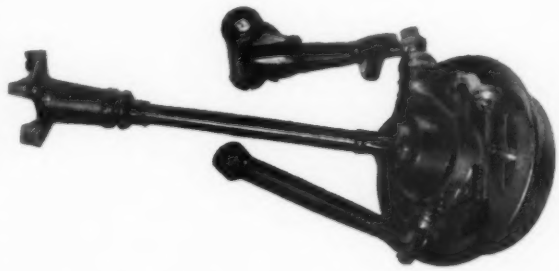
At each end of the Armstrong ⅞ in bore telescopic dampers is a rubber bushed eye; the bushes at the lower ends are carried on studs screwed into the suspension upper links, while the upper ones are mounted on pins projecting from brackets attached through the wing valances to the transverse member that sits on the turret brackets of the subframe. The dampers are installed ahead of the springs, vertically as viewed in side elevation, but with a slight outward inclination that brings the upper ends outboard of the wing valances just mentioned. Orifice type damping is employed, so relatively little resistance is given to slow wheel movements, but a progressive build-up occurs with increasing speed of deflection. There is no doubt that the combination of the rubber springs and orifice type damping has endowed these vehicles with riding qualities that are unusually good in relation to the short wheelbase.

The roll centre height is approximately 5 in when the vehicle is normally laden, a relatively high figure for this type of suspension particularly since the wheels are of below average size. However, it was deliberately chosen in relation to other design considerations, with a view to obtaining the best possible cornering characteristics. The other suspension data are as follows: camber angle of front wheels at static load position, 2 deg; maximum change of camber angle with suspension movement, 2½ deg; maximum variation of track, 2⅛ in; instantaneous periodicity at static load position, 83 c/min.

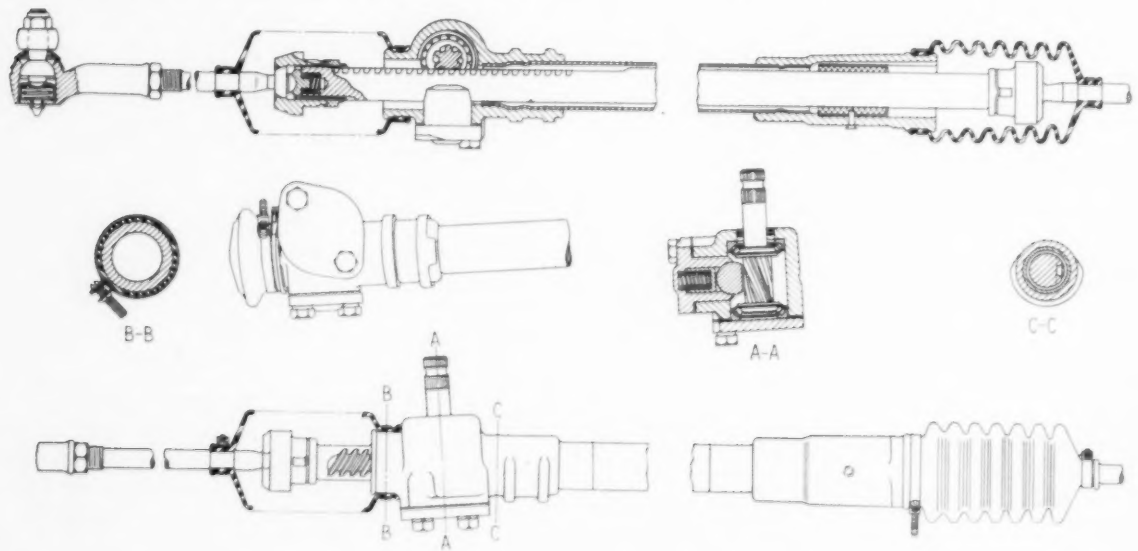
It is, perhaps, apposite to mention at this stage that, in spite of the concentration of the major mechanical units at the front, these vehicles are not unduly nose heavy. The unladen weight distribution of the car is 64:36 per cent as between the front and rear wheels. With the driver only on board, the ratio becomes 63:37; with three passengers and 50 lb of luggage it is 57:43 and at full load it is 53:47. In the case of the estate car version and the van, the corresponding ratios for the unladen, driver only and fully laden conditions are 58:42, 55:45 and 48:52. It could, therefore, be expected that the two longer-wheelbase models would show a slight over-steering tendency at full load.

Steering mechanism

The Cam Gears rack and pinion steering unit is of generally similar design to that of the Morris Minor and Austin-Healey Sprite cars, but is naturally of lighter construction. As was mentioned in Part I of this article, it is mounted on the body structure; the ends of the rack are connected to the steering arms by the usual ball-jointed tie-rods. The arms project behind the steering swivel members and the linkage is such as to provide a geometry approximating to Ackermann over most of the steering range. As is often necessary with rack and pinion steering, the column has an oblique inclination, its lower end being nearer to the car's longitudinal axis than is the upper end. This obliquity, of course, is to bring the pinion far enough



Below: A simple rack and pinion steering mechanism is employed. The aluminium housings for the pinion and the rack bearing are cast on to the steel rack tube, and the helix angle of the pinion is greater than usual. Above: Each transverse link of the front suspension is a steel forging. The malleable iron swivel member carries the back plate of the brake assembly, which has leading and trailing shoes



inboard for the tie-rods to be of adequate length in relation to the suspension geometry. The longitudinal rake of the column is steeper than is now customary, but the combination of small wheels and the rearward mounting of the rack makes this unavoidable without undue complication of the layout.

Comprising the housing of the unit is a 1⅛ in diameter, 16 s.w.g. steel tube, on to the ends of which are cast the aluminium pinion casing and the carrier for the rack bearing. The tube is corrugated near its extremities to provide a positive key for the castings. Two U-bolts, one clamping round each casting, secure the unit to the seating bracket under the toe-board: that at the pinion end of the assembly fits between two circumferential half-ribs on the pinion casing, as a means of transverse location.

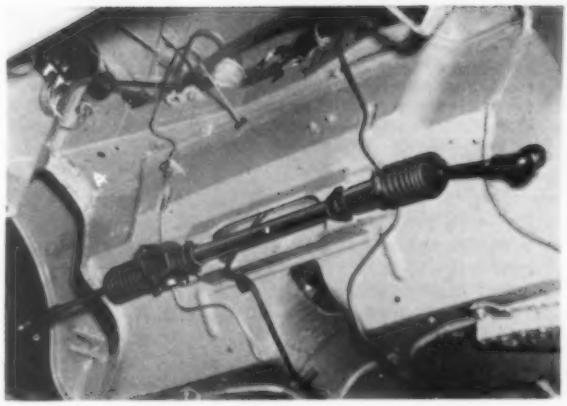
The rack is of En.8C steel. It is 23⅜ in long between the centres of the ball-joints. The material of the pinion—which is situated on top of the rack—and of its integral shaft is En.36A. At the upper end of the shaft there are serrations for the attachment of the column. The working portion of the rack has 22 teeth, and the pinion has six. To ensure a smooth action and to reduce the degree of reversibility, the pinion teeth have the relatively large helix angle of 34 deg 5 min; since the angle between the pinion axis and that of the rack is 85 deg 55 min, the teeth on the rack have a transverse angle of 30 deg. The N.D.P. of the pinion teeth is 14·514.

Supporting the pinion in the casing are two ball bearings of the cup and cone type. Each cone is pressed on to the pinion member, and the cup of the upper bearing is pressed into the casing, which is open at its lower end. The lower cone is a push fit in the casing and bears against the plate that covers the opening; shims are fitted between the plate and the casing to give the desired 0·002 in nip on the bearings. In the upper wall of the casing is fitted a lip type oil seal, which bears on the pinion shaft.

At its end remote from the pinion, the rack is carried in a steel backed felt bearing, which is pressed into the carrier and is secured by a self-tapping screw. In the pinion casing, beneath the rack, is the rack support yoke, which is in the form of a pedestal with a flange at the bottom and a half-bearing at the top. The yoke is a hot brass stamping, to B.S.218, and its pedestal portion is a push fit in a second opening in the pinion casing; the flange is secured to the casing by two set-screws. At the bottom of the half-bearing, a blind hole is drilled axially into the pedestal to house the phosphor-bronze damper pad and its spring. The pad is in the form of a piston embracing most of the spring, which is made of square-section wire.

Adjustment of the mesh of the rack with the pinion is effected by fitting shims between the flange of the pedestal and the face of the casing. The adjustment is made with the rack in the straight ahead position and is correct when rack deflection, caused by turning the steering wheel, imparts a movement of 0·002 in to the damper pad. A thrust of approximately 80 lb is exerted by the damper spring when correctly adjusted.

The ball-joints at the ends of the rack are of the double-diameter type already described in connection with the front suspension. Each ball is integral with its tie-rod, which is made of En.352 steel. The seating for the large-diameter part of the ball is machined directly in a housing sleeve, which is screwed on to the rack and is secured by a lock nut. That for the other part of the ball is in a separate component of approximately mushroom shape. Its stem enters an axial hole in the end of the rack, in which it is embraced by a helical spring of square-section wire; the purpose of the spring is, of course, to compensate for any wear of the joints, which are lubricated by the SAE90 hypoid oil with which the unit is charged. Enclosing each ball-joint is a convoluted bellows of synthetic rubber, which is clipped to the rack housing and to the tie-rod.



The steering unit is mounted on the toeboard, thus facilitating the removal of the complete front assembly, comprising the sub-frame, engine-transmission unit and suspension. On the dash is the cross member to which the turret brackets of the sub-frame are attached

Each tie-rod assembly is 8⅛ in long between the centres of its ball-joints. At the outboard ends of the tie-rods are ball-joints of the orthodox single-diameter type; each ball seating is integral with the housing, which is a steel stamping, screwed on to the tie-rod and secured by a lock-nut. The ball pins have the usual taper mounting arrangement in the ends of the steering arms. A synthetic rubber gaiter encloses each joint, to retain the lubricant and exclude dirt.

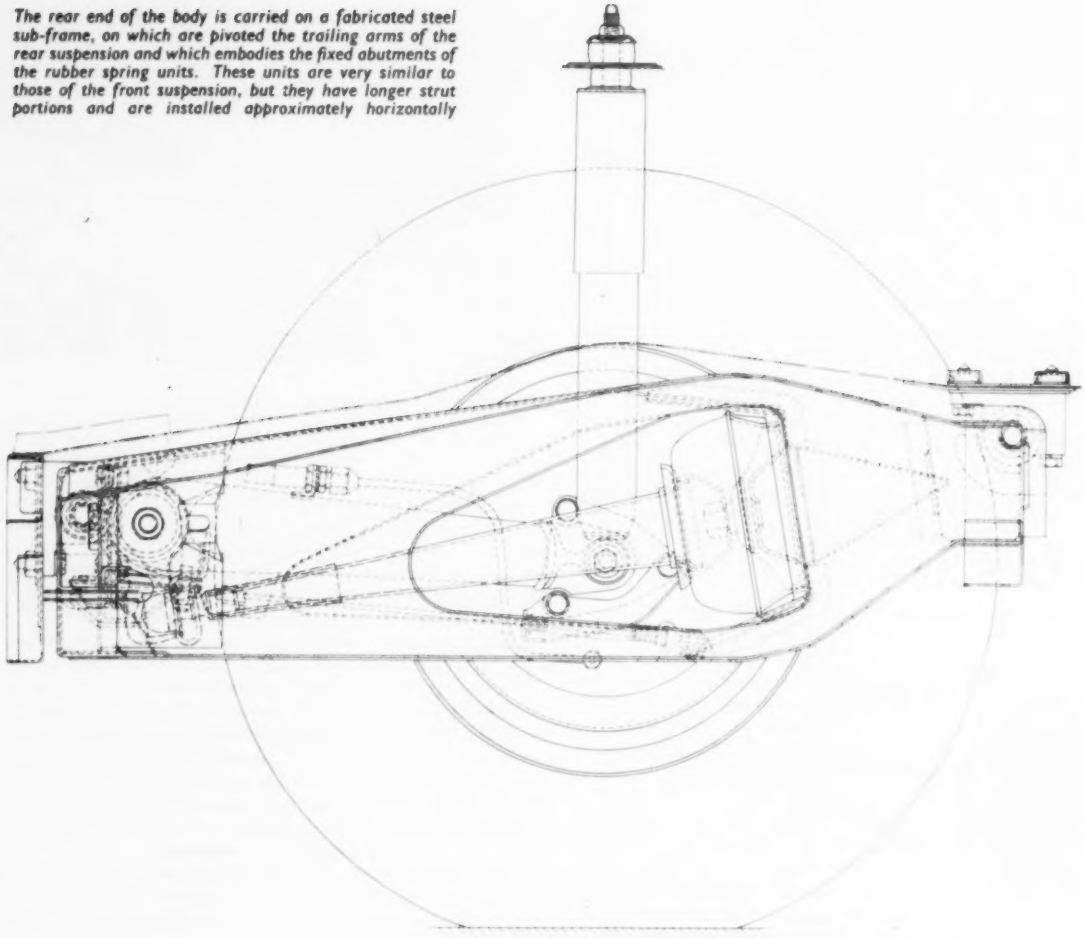
The maximum locks provided by the steering mechanism are 30½ deg inner and 26½ deg outer, at normal riding height. Full lock is reached in each direction when the ball-joint housing at the appropriate end of the rack comes into contact with the end casting. The total steering angularity is, of course, kept within the operating range of the Birfield outboard universal joints of the half-shafts. The turning circle of the car in each direction is 29 ft 6 in between kerbs, about average for a vehicle of 6 ft 8 in wheelbase and a front track of just under 4 ft. Because of the longer wheelbase, both the estate car and the van have a greater turning circle, of 31 ft. From lock to lock, 2½ turns of the steering wheel are required.

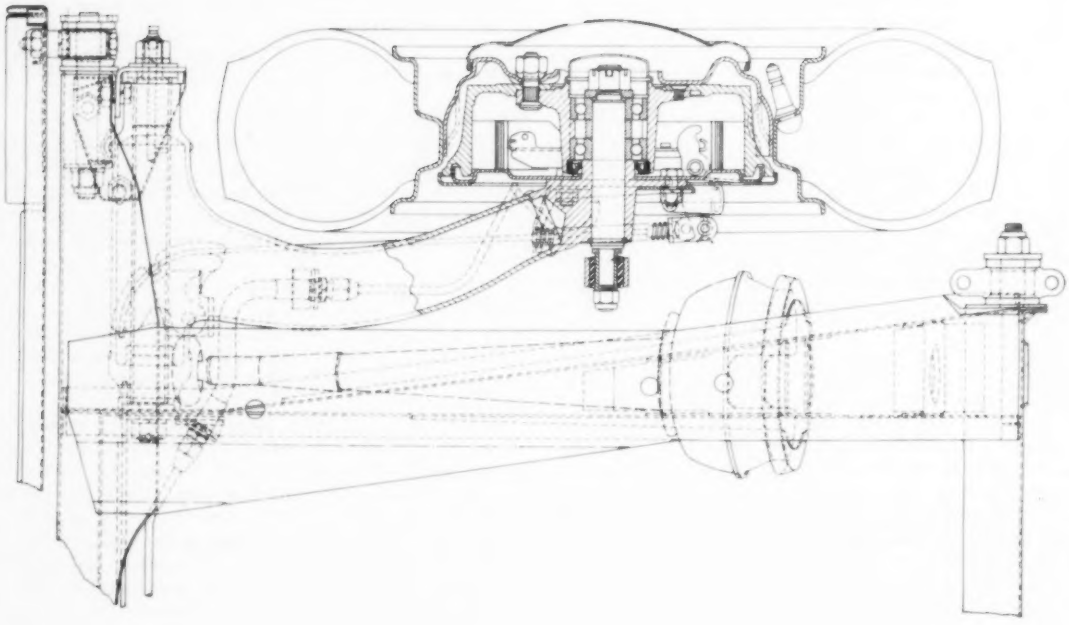
Rear sub-frame and suspension

Most of the steel pressings of which the rear sub-frame is made are of 16 or 18 s.w.g. material. As viewed in plan, the unit is a trapezium, the base of which is to the rear. The front cross member is basically of channel section and projects beyond the slightly inclined longitudinal members, between which it is closed, to form a box section, by a second channel welded within it; both channels face rearward. At its extremities, the cross member is reinforced for the attachment to the body structure, to which reference will be made later, and has bolted-on brackets to carry the outer ends of the pivot bolts for the trailing arms of the rear suspension.

Considerations of space made necessary an almost horizontal, longitudinal disposition of the rubber spring units: by mounting the units partially within the longitudinal members of the sub-frame, a very neat installation has been achieved. In essence, these members are of flanged box section, and they are deepest towards the rear. At the rear of each member, the side webs are separated by almost the full width of the flanges, or top and bottom plates, but they converge towards the front of the sub-frame and are side-by-side for the leading 4 in; their front ends are turned outward

The rear end of the body is carried on a fabricated steel sub-frame, on which are pivoted the trailing arms of the rear suspension and which embodies the fixed abutments of the rubber spring units. These units are very similar to those of the front suspension, but they have longer strut portions and are installed approximately horizontally

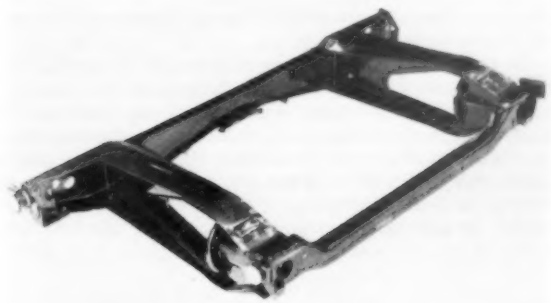




and spot welded to the front cross member. Each web has a shallow flange at top and bottom, spot welded to the main flange plate, and has a large pierced hole of approximately triangular shape to accommodate the spring.

The rear edges of each pair of these holes are bridged by a lipped annular pressing, which is welded in position and forms the seating for the outer cone member of the spring. This seating faces longitudinally forward and slightly downward. Immediately behind it, the depth of the longitudinal member tapers relatively steeply to that of the rear cross member, which is of channel section. To carry the body attachment brackets, the ends of the cross member are closed and reinforced by additional pressings.

In contrast to the rigid attachment of the front sub-frame, rubber is interposed between the rear sub-frame and the body structure. Projecting laterally from each corner of the



Above: The rear sub-frame has the plan form of a trapezium and is attached at its four corners to the body shell; rubber bushes are embodied in the attachment brackets. Right: This illustration shows the mounting of the trailing arms and the neat installation of the springs, which are actuated by short, integral levers on the arms

sub-frame are special pins, on which are carried rubber bushed brackets. These brackets are mild steel forgings, and each of the leading pair is secured by two horizontal bolts to the box section cross member of the body, below the rear seat. In the case of the rear brackets, the corresponding bolts pass vertically through a transverse stiffener at the back of the boot floor.

Each rubber bush is of the transversely split type, with a flange at the outboard end of each half. At the front of the sub-frame, the pins carrying the brackets have plain circular heads at their outboard ends. On to each pin is placed one half-bush, the attachment bracket and then the other half-bush, after which the pin is assembled on to the sub-frame and secured by a stiff-nut at its inboard end. A shoulder on the pin prevents the rubber from being excessively compressed when the nut is tightened. The arrangement at the rear differs in that each pin is secured in the sub-frame by welding, and the stiff-nut retaining the bracket and the half-bushes is at the outboard end. Again the pin has a shoulder to limit the nip of the bushes; this resilient mounting of the brackets obviates any risk of distortion on tightening the bolts that hold the attachment brackets to the body.

The trailing arms of the rear suspension each have the plan form of a T with a curved leg; the bar of the T is the pivot portion of the arm. Each arm is fabricated from two 10 s.w.g. steel pressings and three steel forgings, two of which house the pivot bearings and the third carries the stub axle and the brake back-plate. One pressing forms the upper half of the arm, and the other forms the lower half, the two being seam welded together. Since the two are mirror images, both arms are basically similar, though one is inverted relative to the other, and there is the obvious opposite handing of the spring abutment arms—which are integral with the inboard bearing housings—as well as of the anchorage brackets for the brake pipes and the fairleads for the handbrake cables. The three forgings are butt welded to the pressings.

For rigidity, there are fillet radii between the pivot and lever portions of the arm: the section of the lever portion changes progressively from a flattened ellipse, immediately behind the pivot, to circular at the rear end. As viewed in plan, the curvature of the arm is first inward from the pivot and then outward towards the wheel. In this way the arm is brought clear of the tyre without the plane of the wheel overhanging the outer of the two pivot bearings. The effective length of the arm, between the axes of the pivot and the stub-axle, is $11\frac{3}{4}$ in.

Lead-bronze bushes comprise the pivot bearings, which are situated 7 in apart; this wide spacing provides adequate resistance to the moments resulting from lateral loadings during cornering. Although the welding of both the bearing housings to one arm is done in a single jig, the bearings are line reamed after being pressed in. The nominal bore of the bearings is $1\frac{1}{2}$ in, and their length is $1\frac{1}{4}$ in.

The material of each pivot spindle is En.8 steel. At both ends of the spindle the diameter is reduced to $\frac{1}{2}$ in, where it is threaded for the retaining nuts. At its inboard end the spindle is carried in the double web of the appropriate longitudinal member of the sub-frame, which is locally reinforced for this purpose; as has previously been stated, the outboard support comprises a bracket bolted to the end of the front cross member. Hardened steel thrust washers are fitted between the shoulders on the spindle and the faces of the sub-frame and bracket, and are clamped in position when the retaining nuts are tightened. Spring washers are fitted under the nuts.

Lubrication of the pivot bearings is effected through a grease nipple screwed into the outboard end of each spindle. From the nipple, an axial and a radial hole convey the grease to the annular space within a distance tube between the bushes. Sealing at each end of the pivot is carried out in a similar manner to that of the upper links of the front suspension.

The spring abutment arm, already mentioned as being integral with the inboard bearing housing, projects downward, and near its lower end is a blind hole for the nylon seating of the spring unit. To give the desired 5:1 leverage, the effective length of this arm, from the pivot axis to that of the hole, is 2·35 in. The seating and the ball-end of the

spring unit are identical with those employed in the front suspension, as is the method of sealing.

Although the spring units themselves are basically the same as those of the front suspension, there are two differences to suit the dissimilar installation: the strut portion of each unit is longer, and a progressive bump stop is embodied behind the rubber element, since an external stop would be impracticable with the layout adopted. This stop is in the form of a dished pressing situated within the outer cone member and secured to the spring seating. The wheel rate at the static load position is about 98 lb/in, and it rises on bump to give a maximum wheel load of approximately 1,100 lb.

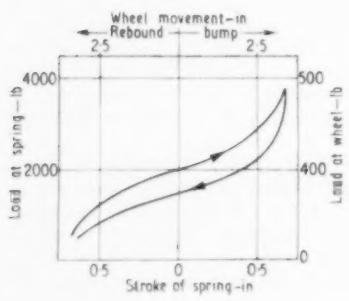
Over most of their length the stub axles, which are of a 70 ton steel, have a diameter of $\frac{7}{8}$ in, but at their inboard ends this is reduced to $\frac{3}{4}$ in to form anchorage pins for the lower ends of the dampers. Only in respect of their length and upper attachments do the dampers differ from those installed at the front: a rubber sandwich type of fitting, connecting to the top of the wheel arch structure, is employed instead of a rubber bushed eye and pin. At the outboard end of each stub axle, the diameter is reduced to $\frac{5}{8}$ in; the extremity is threaded to accept the hub securing nut.

Each stub axle is pressed into its hole in the forging at the rear end of the arm. It is located by a square-section circlip, sprung into a groove at the inboard end of the larger diameter portion of the axle. A shallow counterbore in the forging forms a register for the circlip and prevents it from coming adrift. Towards the outboard end of the forging is the integral attachment flange for the brake back-plate, which is located by a spigot and secured to the flange by three $\frac{7}{16}$ in diameter bolts.

In spite of the basic differences between the front and rear hub assemblies, the accompanying illustrations of them give evidence of a common design technique. Because of its less arduous duty, though, the rear assembly is the lighter of the two. The two ball bearings, which again are of the angular-contact type, are smaller and have an overall diameter, bore and width of 2 in, $\frac{7}{8}$ in and $\frac{9}{16}$ in respectively. Their outer races are pressed into the B.S.310/3 malleable iron hub casting, and are separated by a circumferential rib, whereas the inner races are pressed on to the stub axle and alternate with spacer rings; the inboard inner race seats against the end face of the spigot on the forging, to which the complete assembly is clamped by means of a castellated nut.

The inboard end of the hub member is counterbored to receive a lip type oil seal, which bears on the spigot of the stub axle carrier. At the outboard end is a second counterbore, into which is pushed a grease retaining cap. As in the case of the front hub bearings, no provision is made for routine greasing, because of the risk of the lubricant's reaching the brakes. It is considered that repacking the bearings every 6,000 miles is quite sufficient to ensure adequate lubrication.

Reference has already been made to the fact that, as with any trailing arm suspension layout, the roll centre is at ground level. The instantaneous periodicity of the suspension is 100 c/min at the static load position. As a means of slightly reducing the basic understeering characteristics of the car, the rear wheels have a small positive camber, of 1 deg, obtained by inclining the axis of each stub axle in relation to that of the pivot of the trailing arm.

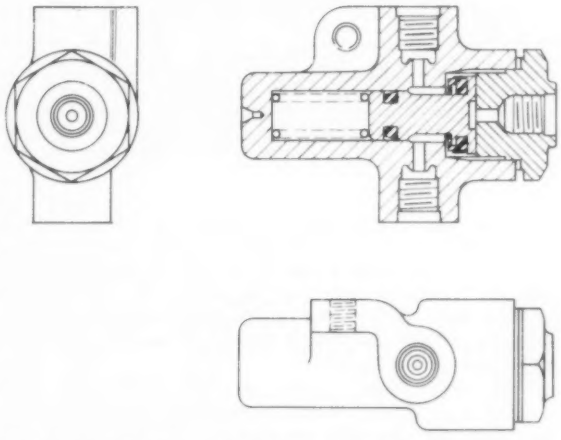


A graph of the load-deflection characteristics of a typical Moulton spring. The manner in which the spring rate varies is illustrated, as too is the high hysteresis

Brakes and wheels

The Lockheed hydraulic braking system is largely orthodox, but several features of it are worthy of comment. First, by virtue of the use of 7 in diameter drums and $1\frac{1}{4}$ in wide linings a relatively large total swept area, of 110 in^2, has been obtained. This is no less than 196 in^2/ton of the kerb weight, and so all four assemblies have leading and trailing shoes: since the high shoe factor of two-leading-shoe front brakes was unnecessary in terms of braking power, it was thought preferable to take advantage of the better thermal stability of the other type. Moreover, the resultant degree of standardization of components has an obvious economic advantage.

On the earlier prototypes, some loss of braking efficiency was experienced owing to water finding its way into the brake drums. This trouble was cured by the adoption of a double labyrinth arrangement: not only has the back-plate the usual lip that partially enters a circumferential groove in the rim of the brake drum, but to it is spot welded an annular pressing that also is lipped. The second lip is of slightly larger diameter than the rim, and it overlaps this.



The hydraulic effort applied to the rear brakes is limited by this valve, the piston of which moves against its spring when the pedal load exceeds 40 lb, thereby isolating the inlet port from the two outlet ports

For the brake drums, a cast iron to B.S.1452 Grade 17 is employed. It is noticeable from the illustrations that the drum width is sufficient to accommodate shoes $\frac{1}{4}$ in wider than standard, so a 20 per cent increase in swept area could be obtained if subsequent developments were to make this desirable. The production lining material is Mintex M32.

A Lockheed pressure limiting valve is installed in the hydraulic line to the rear brakes. The purpose of this device, which is featured also on the Austin A.99 and Wolseley 6/99 cars, is to restrict the effort that can be applied to the rear brakes, thus reducing the likelihood of their locking when the car is lightly laden. This valve, which comes into action when the pedal load exceeds 40 lb, is of the simple spring-loaded piston type; when the spring thrust is overcome by the hydraulic pressure, the piston

moves and its rubber sealing ring comes into contact with a seating face in the body of the valve, thereby shutting off the rear brakes from any further increase of pressure. When the valve is not operating, the ratio of the distribution of braking effort between the front and rear wheels is 63:38 per cent.

The pressed steel wheels have a four-stud attachment to the hubs, to which the brake drums are secured by two countersunk screws, in the conventional manner. Instead of being screwed into the hub, the studs are pressed in; they have serrated shanks, to ensure that they do not rotate, and shallow countersunk heads that seat on similarly countersunk faces in the hub. Because the rims have a diameter of only 10 in, they closely encircle the brake drums, and the nave portions of the wheels are relatively shallow.

In certain quarters, the advisability of adopting such small wheels has been questioned on the grounds that the tyres would drop more readily into minor concavities of the road surface, that a hitherto non-standard tyre size was required, and that wear would be undesirably rapid. The reply of B.M.C. to these arguments is that the small wheels look right on so small a car; that they enable a small turning circle to be obtained without excessive encroachment of the wheel arches into the foot space; that the reduction in unsprung weight more than compensates, in terms of the effectiveness of the suspension, for the slightly increased tendency to vertical movement of the wheel; that the Dunlop Rubber Co. is producing a sufficient quantity of the tyres to meet all requirements; and that the ratio of expected mileage to replacement cost is certainly not inferior to that of the larger tyres that are fitted to other small cars.

The wheels have a rim width of 3·5 in and are fitted with 5·20—10 in tubeless tyres. In the case of the car, the recommended tyre pressures are 24 lb/in² front and 22 lb/in² rear. Because of the greater rear-end weight of the estate car and van versions, higher pressures are necessary in the rear tyres: they are 24 lb/in² for normal running, and 28 lb/in² for those occasions when a full load is carried.

SPECIFICATION

SUB-FRAMES: Front and rear sub-frames fabricated from steel pressings and bolted to body structure. Front sub-frame carries engine-transmission unit and provides mountings for front suspension links and springs; rear sub-frame provides mountings for rear suspension arms and springs. Bare weight of front and rear units, 35 and 37 lb respectively.

SUSPENSION: Front, double transverse link type, with semi-trailing radius rods to lower links. Ball-joints between links and steering swivel members. Vertically mounted, variable-rate rubber springs and separate telescopic dampers. Rear, independent by means of trailing arms. Variable-rate rubber springs, mounted nearly horizontally, and separate telescopic dampers.

STEERING: Rack and pinion. 2½ turns of steering wheel from lock to lock. Turning circles: car, 29 ft 6 in; estate car and van, 31 ft.

BRAKES: Hydraulic, with leading and trailing shoes at both front and rear. Drum diameter, 7 in; lining width, 1¼ in. Total swept area, 110 in². Mechanically actuated hand brake.

WHEELS: Steel disc type with four-stud attachment and 3·5 in wide rims. Tubeless tyres, 5.20—10 in.

Alloy Steels

A BOOK recently issued under the title "Alloy Steels", is obtainable from Samuel Fox and Co. Ltd, of Stocksbridge, Sheffield. Although it includes a very small section on the Fox range of steels, the bulk of this book comprises extremely useful data on many of the commonly used constructional alloy steels. The presentation of this data is such as to be of the greatest possible use to metallurgists.

Information is given on the newer concepts of hardenability, isothermal and continuous cooling transformation, as well as on the effects of alloy composition mass and heat treatment on mechanical properties. This work is indeed a most comprehensive collection of technical data on a selection of the British Standard alloy steels. It is obtainable from Samuel Fox and Co. Ltd, at a price of 2 gns.

Avon High-Hysteresis Tyre

IT is now widely appreciated that tyre adhesion on wet road surfaces can be considerably enhanced by the use of a high-hysteresis rubber compound. However, the amount of heat generated internally is inevitably greater for a high-hysteresis material than for one of low hysteresis. Consequently, if high-hysteresis rubber is used for the casing as well as the actual tread of the tyre, it has been found necessary to impose a restriction on the sustained maximum speed of operation, to avoid the risk of failure owing to overheating.

In this context, the Turbospeed Mark II tyre, recently introduced by The Avon India Rubber Co. Ltd, represents a considerable advance in tyre technology since, for all practical purposes, no such speed restrictions apply. The makers do, in fact, recommend that the maximum sustained speed should not exceed 125 m.p.h, but 140 m.p.h. is permissible intermittently, for periods of up to 10 minutes, and there are very few roads in the world where these limitations could be exceeded. It is noteworthy that Aston Martin Lagonda Ltd. has already approved the Turbospeed Mark II tyre, which is fitted to the latest Aston Martin DB4 cars.

To keep heat generation down to the desired level, a laminated construction has been adopted: since the major deformations—and hence the greatest potential source of heat—occur in the side walls and carcase, only the tread portion of the tyre is of high-hysteresis rubber, which overlays a continuous, bead-to-bead casing of normal, low-hysteresis material. A perfect bond between the two layers is clearly essential, and it is obtained by extruding the layers together, in a new two-fold process developed by Avon India especially for this application.

Cross-section of the new Avon Turbospeed Mark II tyre, showing how a tread of high-hysteresis rubber is combined with a casing of a normal resilient material

Both the section and the tread pattern of the tyre are similar to those of the Turbospeed Mark I cover, production of which has now ceased. The Mark II tyre is made in seven sizes, ranging from 5·20-13 in to 7·10-15 in; it is to the credit of the manufacturers that they have not found it necessary to make any increase in the selling prices of this range of covers relative to those of the earlier series.

Deceleration Measurement

A Survey of the Fundamental Principles of the Instruments Available, and Factors Affecting their Application

By P. R. J. HARDING

IN automotive brake testing, as is well known, the deceleration of a vehicle is measured either indirectly by means of a stopping distance recorder or directly with a decelerometer. The former method is sometimes used for truck brake testing and less frequently for passenger car work, whilst the latter is used almost universally for brake and brake lining development and proving tests. Because of the wide use of indicating decelerometers, a survey of the general theory of such instruments, with reference to the effects of vehicle tilt and braking on gradients, will be of value. The effect of vibrations and damping on the accuracy and sensitivity of decelerometers has not been included in the survey, because of previous papers[1] on this subject.

Types of decelerometer

Instruments marketed and in use as decelerometers have been referred to in detail in technical pamphlets published by their manufacturers and in papers[2] on brake testing and road research. All these instruments may be divided into categories, under the following headings:

a. Fluid in U-tube types
b. Pendulum mechanisms
c. Spring-mass, or inertia systems.

The first two are the most common because their simplicity and compactness enable them to be readily fitted to and used in road vehicles. They are also relatively inexpensive. The third type, being generally more suitable for incorporating in a recording system, is used more for research purposes than for general brake testing.

Fluid in U-tube

This type of decelerometer is the most simple in construction, consisting of a closed or continuous glass U-tube partly filled with liquid. Its front, or scale, tube is of small bore and can be calibrated in cm/sec², ft/sec² or percentage g deceleration. On the other hand the rear tube, or reservoir, is of relatively large bore, to give adequate sensitivity. The base tube, usually of similar bore to the scale tube, is sufficiently long to give a convenient scale size. Damping can be obtained by means of either a constriction in the base tube or by use of fluid with suitable viscosity, or a combination of both.

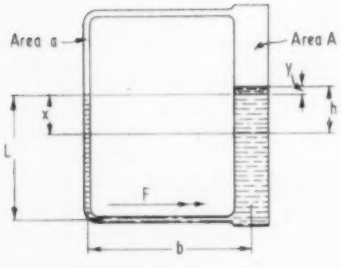


Fig. 1 left: Diagram of U-tube type decelerometer positioned with its base tube horizontal

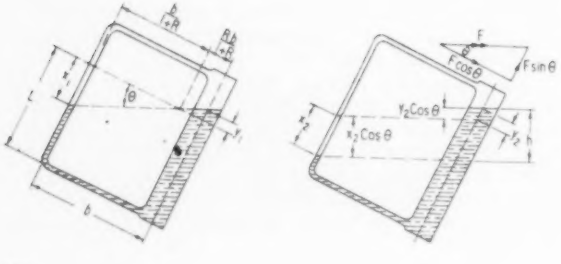


Fig. 2, below: If the U-tube is tilted at some angle θ, the fluid levels are as in this illustration: a, left, under static conditions; b, right, under the influence of a vehicle deceleration force F

With the U-tube positioned so that the base tube is level, Fig. 1, the height of the liquid under static conditions is L, measured from the base tube. This gives the position of zero deceleration on the scale. Owing to capillary action, the height of fluid in the scale tube exceeds that in the reservoir, but this has virtually no effect on the use of the U-tube and will be ignored therefore in the following.

The base tube length is b and the ratio of the cross sectional area of the scale tube to that of the reservoir is such that $a/A = R$. Under an imposed deceleration force F, the fluid column height in the scale tube, will decrease by x to give a scale reading, whilst the reservoir level will rise an amount y such that:

$$x + y = h \text{ where } h \text{ is the fluid head.}$$

$$\text{The ratio } \frac{y}{x} = \frac{a}{A} = R$$

If W is the fluid density and D is the deceleration ratio produced by force F that is, D = deceleration g, the pressures due to h and F can be equated:

$$\frac{Awh}{A} = \frac{awbD}{a}$$

$$\text{hence } x = \frac{b}{1+R}D \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 1$$

The quantity $b/(1+R)$ is a constant depending on the dimensions of the U-tube. Therefore the scale movement x is $\propto$ deceleration, so the instrument has a linear scale.

In the case of a Bennett-Feragen 4 in U-tube decelerometer, the ratio R is approximately 0·034 : 1 and the base tube length is 4 in. The instrument constant $C = b/(1+R)$ is 3·87, that is, the scale length equivalent to g deceleration is 3·87 in.

However, in the general case, owing to either faulty zero setting of the instrument or tilt of the vehicle body relative to the road surface, the U-tube may be tilted at some angle θ to the line of action of the deceleration force. Under static conditions, Fig. 2a, the scale movement x_1 is due solely to the tilt of the U-tube.

$$x_1 = \frac{b}{1+R}\tan\theta \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 2$$

When the U-tube is subjected to a deceleration force F, Fig. 2b, the fluid head h will comprise a drop in level of the fluid in the scale tube and a rise in level in the reservoir. The scale movement $x_2 = h/\cos\theta - y_2$ and $x_2/y_2 = 1/R$. For equilibrium, the fluid pressure due to the deceleration force must be balanced by the fluid head h:

$$\frac{Awh}{A} = \frac{Aw\,(L + y_1 + y_2)D\sin\theta}{A} + \frac{aWbD\cos\theta}{a}$$

$$\frac{aw\,(L - x_1 - x_2)D\sin\theta}{a}$$

Substitutions may be made for L, h, y_1, y_2 and x_1 in terms of x_2, θ, R and b:

$$x_2 = \frac{bD}{1+R}\left(\frac{1 + \tan^2\theta}{1 - D\tan\theta}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots 3$$

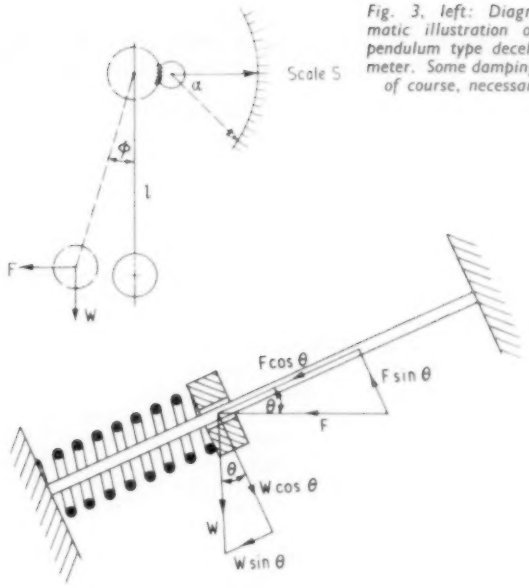


Fig. 3, left: Diagrammatic illustration of a pendulum type decelerometer. Some damping is, of course, necessary

The actual scale reading will be given by adding x_1 and x_2. With an instrument calibrated so that the constant $C = b/1 + R$ is equivalent to g deceleration the scale reading will be S:

$$S = C\left(\frac{D + \tan\theta}{1 - D\tan\theta}\right) \quad \ldots\ldots 4$$

Note that if the scale is calibrated in ft/sec², $C = 32{\cdot}2$, and if calibrated in per cent g, $C = 100$. For any given deceleration D and tilt angle θ, the percentage error in scale reading will be:

$$E = \frac{100\,(1 + D^2)\tan\theta}{D\,(1 - D\tan\theta)} \text{ per cent} \quad \ldots\ldots 5$$

Pendulum type

Constructionally this type of decelerometer consists basically of a small compound pendulum, which is suitably geared to a moving scale or to a moving pointer. Damping is effected by means of fluid. An instrument of this type is represented diagrammatically in Fig. 3, in which the compound pendulum is regarded as a simple pendulum of weight W having effective length l.

A deceleration force F will produce a deflection of the pendulum ϕ, and the pointer or scale will be deflected through angle α, such that α/ϕ = gear ratio.

For equilibrium:

$$Fl\cos\phi = Wl\sin\phi$$
$$F = W\tan\phi$$
$$\phi = \tan^{-1}D \quad \ldots\ldots 6$$

Movement of the scale will be $\propto \tan^{-1}D$, and so non linear.

In many instruments the scale angle α is designed to be 360 deg for g deceleration, while the pendulum deflection is 45 deg at this deceleration. In this case, the gear ratio $\alpha/\phi = 360/\tan^{-1}1 = 8$. The scale reading of the pendulum decelerometer will be given by:

$$x = C\tan\phi \quad \ldots\ldots 7$$

Suppose the pendulum decelerometer is tilted through an angle θ. The scale movement under constant velocity conditions of operation will be due solely to the tilt angle:

$$x_1 = C\tan\theta$$

Under the action of a subsequent deceleration force F, there will be an additional scale movement:

$$x_2 = C\tan\phi$$

The total scale movement will be $x_1 + x_2$, and the scale reading of a calibrated instrument will be proportional to the tangent of the total angular deflection of the pendulum from zero:

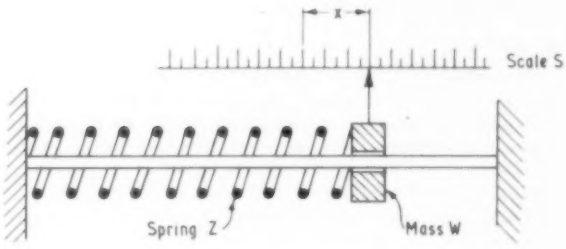
$$S = C\tan(\phi + \theta)$$

Therefore, the scale reading for an instrument tilted at angle θ to the line of action of the decelerating force for any given deceleration D will be:

$$S = C\left(\frac{D + \tan\theta}{1 - D\tan\theta}\right)$$

This is identical to equation (4) of the U-tube decelerometer, so the effect of tilt is the same for both types of instrument.

Fig. 4: Schematic arrangement of the spring mass type of decelerometer: a, below, with the instrument set horizontally; b, left, with it inclined at an angle θ owing to either faulty setting of the instrument or body tilt



Spring-mass type

A decelerometer of this category is shown schematically in Fig. 4a. The mass W is attached to a spring Z so that, depending on the spring rate and mechanism friction, the pointer movement is proportional to the applied deceleration force F.

$$x = CD \quad \ldots\ldots 8$$

The action is usually steadied by attaching a suitable damper to the mass W or, more simply, by immersion or partial immersion of the mechanism in a suitable fluid.

Fig. 5, left: The test vehicle with strips of tape horizontally aligned on the side of the body. Fig. 6, right: Tilting 4 deg at 100 per cent g braking

If the instrument is tilted, the spring will deflect slightly under the influence of the component of the mass W down the plane, Fig. 4b. Under the action of a deceleration force F, the actuating force component will be $F \cos \theta$, giving a total spring deflection proportional to:

$$W \sin \theta + F \cos \theta.$$

The scale reading for a calibrated instrument will be given by:

$$S = C (\sin \theta + D \cos \theta) \quad \ldots\ldots\ldots\ldots\ldots\ldots 9$$

With a tilted instrument, the percentage error in scale reading will be smaller than with the U-tube or pendulum types.

$$E = 100\left(\frac{\sin \theta}{D} + \cos \theta - 1\right) \text{ per cent} \quad \ldots\ldots\ldots 10$$

A tilt angle of 3 deg at 100 per cent g deceleration will give an error of 5 per cent.

Vehicle tilt

One effect of braking a forward moving road vehicle is of course the transference of weight from rear axle to front axle, and the consequent deflections of the springs, tyres and other suspension components causes a tilting of the vehicle body relative to the road surface. The amount of weight transfer, and therefore the angle of tilt, θ, is proportional to the deceleration. It may be reasonably assumed that $\tan \theta \propto D$, although for the relatively small angles involved it is equally true that θ or $\sin \theta \propto D$.

Experimental work has been carried out to measure tilt of a vehicle body at various deceleration rates. The technique used was to photograph various vehicles whilst they were being decelerated at constant rates. Each vehicle was fitted with two pendulum type decelerometers, and strips of suitably coloured tape were fixed in a horizontal line along the side of the body to be photographed.

Initial test runs were made at zero deceleration to establish a datum for measurement of body tilt angle. Next, the vehicles were braked at various deceleration rates, which were maintained at constant values by an experienced brake test driver using one of the pendulum decelerometers. The observer made a more accurate reading of the deceleration from a second decelerometer. Measurements of tilt were made from the film by noting the inclination of the tape strips.

Preliminary experimentation showed that, because of friction effects in vehicle suspensions, different tilt angles may be obtained for similar deceleration rates. It was found that the rate of application of the deceleration force influenced the tilt angle results. For example, if the deceleration was gradually brought up to the required value, the tilt angle was slightly lower than if the deceleration was increased rapidly to reach the target figure. Overshooting of that figure, followed by a reduction in braking to register and maintain the required deceleration resulted in higher tilt angle readings.

The effect of residual tilt due to suspension friction was investigated statically by depressing and raising the front and rear suspensions of various private cars simultaneously and then allowing them to settle slowly to no load positions. Measurements showed residual tilts of between 0·2 and 0·7 deg for these various cars. Dynamic tests gave similar results.

Tests were made to examine the effect of initial speed of the vehicle on tilt angle and, whilst it was found that there was no significant trend in results, it was established that an initial speed of approximately 40 m.p.h. was most satisfactory for tilt measurements by photographic methods. This speed allowed sufficient time to register and maintain the target deceleration figure before running past the camera. A sufficient number of tilt angle measurements was made to enable some conclusion to be drawn as to the relationship between tilt angle and deceleration and the magnitude of the tilt. Illustrations of a stationary car and the same car under high deceleration are shown in Figs. 5 and 6.

The tilt angle-deceleration results for a large passenger car are plotted in Fig. 7. Similar results obtained from a medium sports saloon are shown in Fig. 8, and Fig. 9 gives the results obtained with a 7 ton truck. In all these curves tilt angle is plotted against the deceleration, as indicated by a pendulum type decelerometer. In Fig. 10, these same results are corrected for the effect of tilt angle on deceleration reading and extended to 100 per cent g, to obtain the tilt factor T. The tilt factor T is defined as the tangent of the tilt angle at a true 100 per cent g deceleration. Actual and corrected results obtained from a variety of passenger cars, at a target deceleration figure of 80 per cent g, are shown in Fig. 11.

Practical effects

From (4) the scale reading for either pendulum or U-tube decelerometers subjected to tilt is given by:

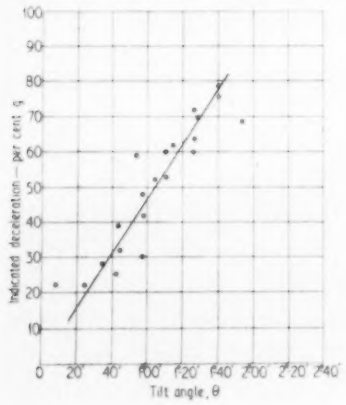
$$S = C\left(\frac{D + \tan \theta}{1 - D \tan \theta}\right)$$

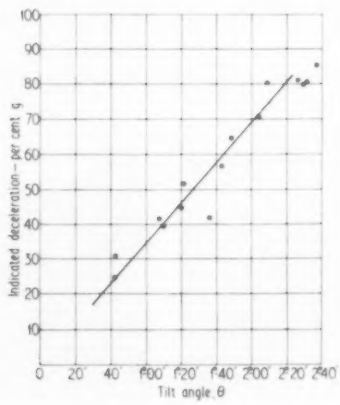
If two such instruments are set in a vehicle at different tilt angles θ_1 and θ_2 the difference in scale reading at any given deceleration will be:

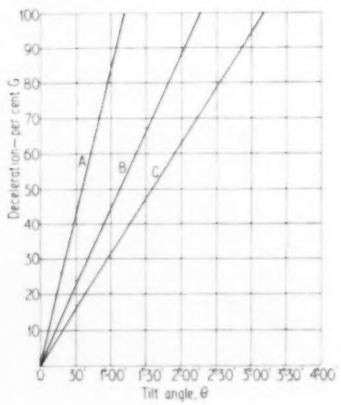
$$d = C\,\frac{(1 + D^2)(\tan \theta_1 - \tan \theta_2)}{(1 - D \tan \theta_1)(1 - D \tan \theta_2)} \quad \ldots\ldots\ldots 11$$

If one instrument is set level so that $\theta_2 = 0$ and if it can be

Fig. 7, left: Tilt angles obtained from pendulum tests on a 2½ litre saloon. Fig. 8, centre: Those from a 1½ litre sports saloon. Fig. 10, right: Corrected tilt factor lines: A 7 ton truck, T=0·02003, I/T=49·93; B family saloon, T=0·03963, I/T=25·23; C sports saloon, T=0·05550, I/T=18·02







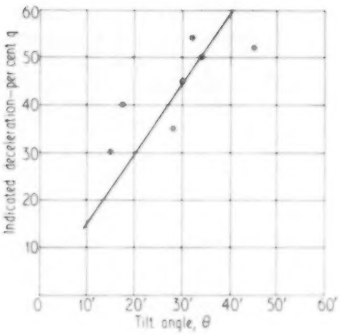
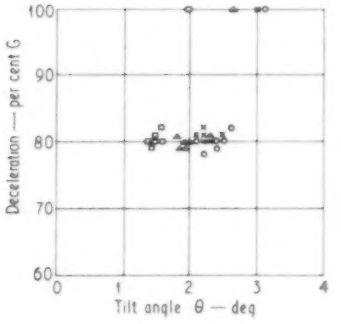


Fig. 9: Tilt angles obtained from pendulum tests on a 7 ton truck



□ 2 litre sports car; △ 2 litre saloon; × 2½ litre saloon; ○ 1½ litre saloon

Fig. 11: Tilt results for various passenger cars. The group of four plots at the top are the average readings corrected to 100 per cent true deceleration, while the group in the centre are the actual readings taken with a pendulum decelerometer

assumed that the vehicle tilt angle is small relative to θ_1.

$$d = \frac{C(1+D^2)\tan\theta_1}{1-D\tan\theta_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots 12$$

Hitherto only positive tilt has been considered, that is, tilt in the same sense as that of vehicle tilt produced by braking a forward travelling vehicle. For negative tilt, θ will be negative.

$$\text{Therefore } d = \frac{-C(1+D^2)\tan\theta_1}{1+D\tan\theta_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots 13$$

Experiments were made using a sports car, with a low centre of gravity and stiff suspension, fitted with pairs of pendulum and U-tube decelerometers. One instrument in each case was set correctly with the vehicle stationary on a level stretch of road, whilst the setting of the second instrument was made to give various degrees of set tilt. Tests were made from 40 m.p.h. and the driver maintained various constant deceleration values by observation of the correctly set decelerometer. The scale readings of both instruments were recorded by the observer.

In the U-tube tests, the set tilt was varied to give stationary readings from 1 ft/sec² to 6 ft/sec² and −1 ft/sec² to −6 ft/sec², and the pendulum decelerometer tests were carried out with set differences of ±5 per cent g to ±20 per cent g. In Figs. 12 and 13 the practical results are shown plotted and compared with the theoretical results obtained by using equations 14 and 15. These results are not corrected to allow for the additional effect due to vehicle tilt, because it was considered that the body tilt with the particular vehicle used would be very small by comparison with the set tilt of the second decelerometer. The tilt angle for this car was in fact slightly less than 1·5 deg at a true deceleration of 80 per cent g. However, correlation between uncorrected practical results and theoretical results is sufficiently good to indicate the general effect of tilt on decelerometer readings.

Dependence of tilt angle on deceleration

The percentage error in scale reading of pendulum and U-tube decelerometers due to tilt is given by equation 5:

$$E = \frac{100(1+D^2)\tan\theta}{D(1-D\tan\theta)} \text{ per cent}$$

In this equation, θ and D are particular values of tilt angle and deceleration. From tilt angle measurements on various road vehicles, it is evident that $\theta \propto D$, and for the small angles involved, it may be assumed that $\tan\theta \propto D$:

$$\tan\theta = TD$$

where T is the tilt factor whose value can be obtained by putting $D=1$. Then:

$$E = \frac{100(1+D^2)}{(1/T)-D^2} \text{ per cent} \quad \ldots\ldots\ldots\ldots\ldots\ldots 14$$

The equation has been expressed in terms of $1/T$ for the convenience of using cotangent tables, and it has been used to plot the family of curves in Fig. 14. These curves show the per cent error in scale reading, for pendulum and U-tube decelerometers. Since they are based on true deceleration they are not convenient to use for correction of decelerometer readings. It can be seen that for a given tilt factor, as D approaches zero, the error $E=T$. If the upper limit value of D is unity then $E=2T/(1-T)$. For a body tilt angle of 3 deg at g deceleration the tilt factor $T=0{\cdot}0524$. In this case, assuming tilt is proportional to deceleration, the error will vary from 5·24 to 11·06 per cent, through the range.

From equation 14, $E = \dfrac{100\,T(1+D^2)}{1-TD^2}$ per cent

If the scale reading S is expressed as a deceleration ratio N,

Fig. 12: Differences between the readings of correctly set and tilted U-tubes. These curves are uncorrected for the vehicle tilt and they are plotted against the indicated values of the deceleration of the vehicle

* – – * *4 in Bennett-Feragen U-tube;* ——— *theoretical lines*

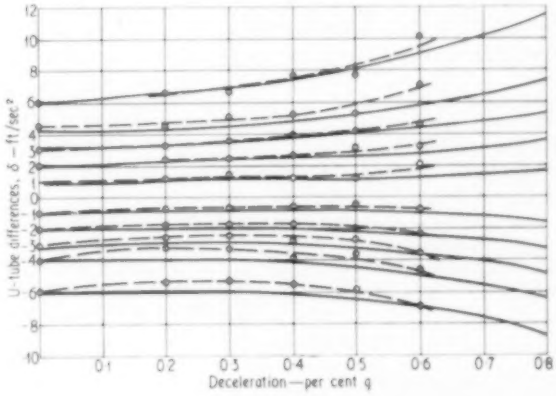
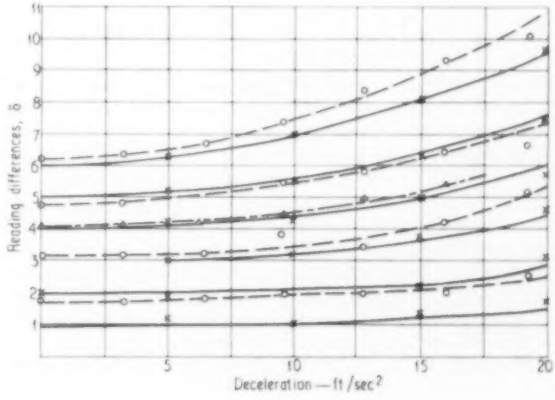


Fig. 13, below: Differences between the readings of U-tube Dynometer and Tapley readings of deceleration compared with the theoretical curves, which are obtained by calculation, using the equations 14 and 15

* – – * *Dynometer and Tapley;* × *U-tube, 4 in Bennett-Feragen;* △ — · — △ *Dynometer on 1 in 8 gradient;* ——— *theoretical lines*



that is, $N = S/C$:

$$E = \frac{N-D}{D} = \frac{T+TD^2}{1-TD^2}$$

This gives rise to a quadratic in D, the solution of which gives an equation for finding the true deceleration:

$$D = \frac{-(1+T)}{2NT} \pm \sqrt{\left(\frac{1+T}{2NT}\right)^2 + \frac{1}{T}} \quad \ldots\ldots\ldots\ldots\ldots 15$$

The scale reading equation is:

$$S = CD\,\frac{(1/T)+1}{(1/T)-D^2} \quad \ldots\ldots\ldots\ldots\ldots 16$$

This has been used to calculate values for Fig. 15, which is a family of correction curves for U-tube and pendulum type decelerometers, for positive tilt angles up to 4 deg. Since they are plotted against indicated deceleration, these curves can be used for correcting actual decelerometer readings, provided the value of T for the vehicle concerned is known.

For the spring-mass decelerometer, assuming $\tan\theta = TD$, from equation 10, it can be established that:

$$E = 100\left(\frac{1+1/T-\sqrt{(1/T)^2+D^2}}{\sqrt{(1/T)^2+D^2}}\right) \text{ per cent} \quad \ldots\ldots 17$$

On the assumption that θ is small, and therefore that θ, $\sin\theta$ and $\tan\theta$ have approximately the same value, an approximate expression for the per cent error can be obtained simply from equation 10:

$$E \text{ approx.} = \frac{100\,\tan\theta}{D} \text{ per cent}$$

If $\tan\theta = TD$, then E approx. $= 100\,T$ per cent. This means that for a given tilt factor the per cent error is approximately

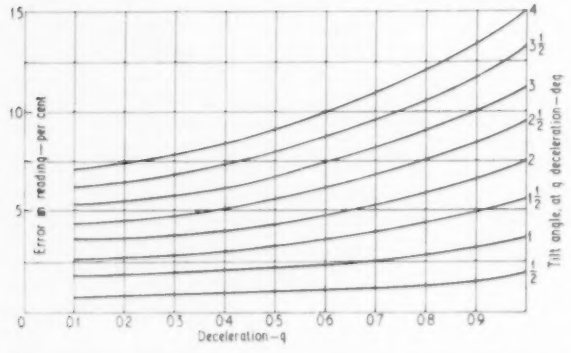
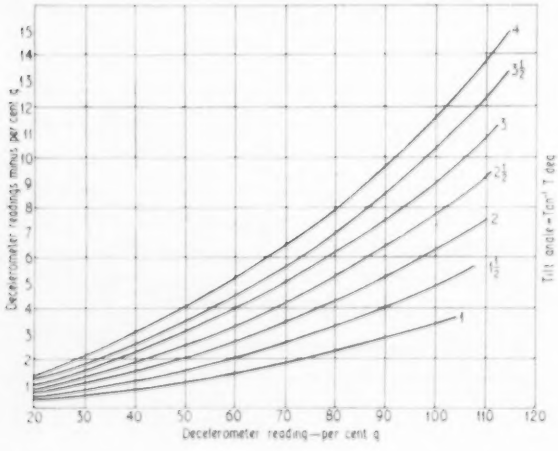


Fig. 14, above: Percentage error in scale reading of pendulum and U-tube types of decelerometer due to tilt where, for small angles, $\tan\theta \propto D$

Fig. 15, below: Correction curves for U-tube and pendulum readings



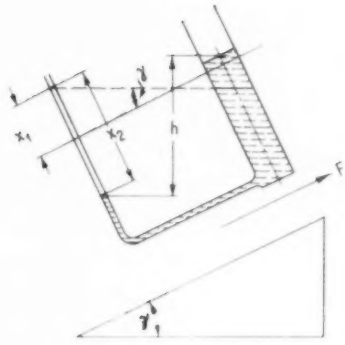


Fig. 16: Diagram representing a U-tube decelerometer on a gradient of angle Y

constant throughout the range of the instrument. For example, a body tilt of 3 deg gives an approximate error of 5·2 per cent at all deceleration values. To calculate deceleration ratios from the readings of a spring-mass decelerometer, the following equation can be used:

$$D = \frac{N}{\sqrt{1+2T+T^2(1-N^2)}} \quad \ldots\ldots\ldots\ldots 18$$

An approximation which is accurate enough for the small angles involved is:

$$D = \frac{N}{\sqrt{1+2T}}$$

Braking on a gradient

In general, most brake test work is carried out on level stretches of road or track. However, hill tests, particularly mountain pass descents, are sufficiently widely used to merit some consideration of the effect of gradient on the readings of decelerometers. The effect of gradient is similar to that of tilt, in that, under stationary or constant velocity conditions, the decelerometer gives a reading that is proportional to the tangent of the angle. However, the effects differ, in that, so far as the influence of the gradient is concerned, the decelerometer remains in the plane of the deceleration force, whereas vehicle tilt causes the decelerometer to be inclined at some angle to the line of action of the deceleration force. It should be noted that tilt also occurs when a vehicle is braked on a gradient, and the degree of tilt depends on the gradient as well as on the actual deceleration attained.

Consider a U-tube decelerometer on a gradient of angle γ, Fig. 16; then, stationary or constant velocity conditions will give rise to a scale reading:

$$x_1 = \pm C\tan\gamma$$

the positive sign representing downhill and the negative sign uphill conditions. The fluid head h will be proportional to the deceleration D and therefore the drop in level in the scale tube will be $\propto D$, so the scale movement is given by:

$$x_2 = \frac{CD}{\cos\gamma}$$

The scale reading Sg will be the algebraic sum of x_1 and x_2

$$Sg = C\left(\frac{D}{\cos\gamma} \pm \tan\gamma\right)$$

$$Sg = C\left(\frac{D \pm \sin\gamma}{\cos\gamma}\right) \quad \ldots\ldots\ldots\ldots\ldots 19$$

It should be noted that D is the actual deceleration of the vehicle, not the brake deceleration. In test work it is usually the brake deceleration that is of main interest, and if this brake deceleration is D_B:

$$D_B = D \pm \sin\gamma \quad \ldots\ldots\ldots\ldots\ldots 20$$

With the pendulum instrument, as in Fig. 17, the scale movement on the gradient, under constant velocity or stationary conditions, is caused by deflection of the pendulum through an angle γ. Therefore, deceleration D will cause a

deflection of the pendulum such that the angle from zero scale position is ϕ, and from equation 7, the reading $Sg = C \tan \phi$.

For equilibrium of the pendulum, $D \cos \phi = \sin (\phi \pm \gamma)$

from which $\tan \phi = \left(\frac{D \pm \sin \gamma}{\cos \gamma} \right)$

and, as in the case of the U-tube, $Sg = C \left(\frac{D \pm \sin \gamma}{\cos \gamma} \right)$

From equation 20, it can be seen that both U-tube and pendulum decelerometers will give a reading on the gradient of $Sg = C\, D_B / \cos \gamma$. Therefore, the percentage reading error due to gradient will be:

$$Eg = 100 \, (\sec \gamma - 1) \text{ per cent} \quad \ldots\ldots\ldots\ldots\ldots 21$$

It can be calculated that the error due to gradient alone is slightly less than 2 per cent on a gradient of 20 per cent, 1 in 5.

With spring-mass decelerometers there is no error due to gradient, because the scale itself is not subject to any modification by angular function. The reading will be given by:

$$Sg = C \, (D \pm \sin \gamma) = C D_B \quad \ldots\ldots\ldots\ldots\ldots 22$$

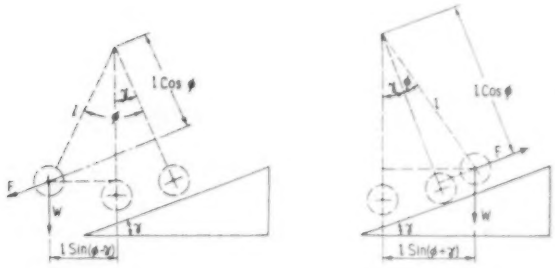


Fig. 17: Pendulum type instrument on upward and downward gradients

Conclusions

With most passenger cars, the magnitude of body tilt angle due to weight transfer under normal braking is not likely to exceed 3 deg. For most practical purposes, passenger car tilt due to braking can be taken as being between 2 deg and 4 deg at 100 per cent g. Commercial vehicles exhibit fairly small tilt angles, of the order of $\frac{1}{2}$ deg to 1 deg for 60 per cent g deceleration.

The tilt of any vehicle may depend on the condition of the suspension and springing, so results can be expected that vary from vehicle to vehicle as well as from type to type. Tilt can be determined by photographic methods but it is probably better to use a suitable gyroscopic instrument.

Because of the effects of vehicle tilt, U-tube and pendulum type decelerometers, when used for car brake tests, may have scale reading errors up to 14 per cent. Spring-mass instruments are less affected by tilt and may exhibit errors up to 5 per cent. When used for testing large commercial vehicles, all three types of decelerometer have tilt errors which, for most practical purposes, are negligible.

It is usual, in passenger car brake testing, to regard results as being comparative, for the purpose of assessing brake and lining performance, and in this case, errors due to tilt and gradient can be ignored on the assumption that similar cars will have similar tilt factors. However, where the deceleration is high and is being used as a performance criterion, the errors may be significant, since the rate of application of the deceleration force and variations in suspension and springing can influence vehicle tilt. Where test work is concerned with detailed analysis of brake performance, decelerometer tilt errors are high enough to demand correction of readings.

Before setting decelerometers it is preferable to allow the vehicle to come to rest, on a level surface, from low speed under the deceleration force due to its own rolling resistance. Care should be taken in setting decelerometers to zero, because a setting error of 1 ft/sec² with a U-tube or pendulum decelerometer may give a total reading error of up to 10 per cent at 20 ft/sec² deceleration.

Although the effects of lateral angularity have not been considered, it is obvious that the errors involved will be similar to those in the case of gradient. It seems advisable, therefore, to ensure that the decelerometer is set to within ± 5 deg of the line of motion and deceleration. The fixing of decelerometers with limited side angle adjustment to very curved windscreens and the temptation to tilt an instrument sideways to aid viewing by the driver or observer should be avoided.

Braking on normal gradients gives rise to very small error with U-tube and pendulum decelerometers and none with spring-mass types, and therefore, for most practical purposes, it can be taken that accelerometer readings obtained on gradients are equivalent to readings obtained on level road from similar braking effort.

From the equations for U-tube and pendulum types it is apparent that a simple and ready calibration procedure can be effected by use of a tilt table. It can also be seen that these types may be used for gradient measurement.

References

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2. H. J. H. STARKS, Dr: "The Braking Performance of Motor Vehicles and Brake Testing", *D.S.I.R. Road Research Laboratory*, Technical paper No. 26, 1953

Michelin 'X' Tyres

TESTS carried out recently by the Michelin Tyre Co. Ltd, and observed by the R.A.C, have shown that appreciable economies in respect of fuel consumption are obtained when Michelin 'X' tyres are used to replace the more conventional multi-ply fabric reinforced types. This economy is obtained by virtue of the radial arrangement of the wall cords, the fact that there are only two plies in the carcass, and the stabilization of the crown in both the circumferential and lateral directions by three additional plies of steel-cord fabric. Because of the radial arrangement of the wall cords, there is no change of angle between those in each ply as the tyre flexes, as there is with biased cord plies, and this, together with the reduction of the number of plies, greatly reduces the generation of heat owing to the working of the rubber between them; furthermore, the steel-cord plies that reinforce the tread help to maintain a flat profile, so there is less distortion of the rubber in this area as it passes into and out of the contact patch, and this again helps to reduce the rolling resistance.

In general, the tests have shown that the larger the tyre the greater is the saving obtained. Several identical pairs of different types of car were tested, one of each pair had Michelin 'X' and the other the original tyre equipment. Those with the 'X' tyres gave extra mileages of 4·4 to 8·5 per cent with equal quantities of fuel. Similar tests with a 7 ton truck showed that the gain could be as much as 15·6 per cent.

BOOKS

Brief Comments on Recent Publications Pertinent to Automobile Engineering

Polythene

Edited by A. Renfrew and P. Morgan

London: ILIFFE BOOKS LTD, Dorset House, Stamford Street, S.E.1. 1960. 10×6¼. 781 pp. Price 165s.

Polythene is now one of the most important basic materials of industry. Its technology is complex and its use widespread. The book under review sets out the basic facts of all aspects of polythene production, properties, handling techniques and major uses in an authoritative appraisal which should be essential to all technologists in the plastics industry and in many other industries where polythene is assuming an important role.

This is the second edition, and it contains approximately 300 pages more than the previous one. In the course of revision, the original text has been largely re-written in order to bring the book in line with the rapid developments in this field, particularly in relation to higher density materials. The contents are as follows: The history of polythene; Manufacturing processes; The theory of ethylene polymerization by free radicals and the kinetics of the high pressure reaction; The theory of ethylene polymerization by ionic growth; The structure of polythene; Oxidation and ageing; Dielectric properties; General mechanical properties; Chemical resistance and physico-chemical properties; Testing and specifications; Low molecular weight polythenes; Radiation treatment of polythene; Ethylene copolymers and polymer blends; Modified polythene; Compounding; Pigmentation; Extrusion problems and screw design; The extrusion of polythene; The extrusion of tube, rod and profile; The extrusion of polythene flat film, tubular film and sheet; Injection moulding; Blow moulding; Centrifugal casting of large diameter pipe; Extrusion coating and laminating; Dip coating; Spray coating; Vacuum forming; Reprocessing of polythene; Cables and other dielectric applications; Film applications; Polythene coated packaging products; Packaging applications of polythene other than film; Polythene pipe for cold water services; Chemical plant; Household goods; and Textiles yarns and fabrics.

Kempe's Engineers Year Book

London: MORGAN BROTHERS (PUBLISHERS) LTD, 28 Essex Street, Strand, W.C.2. 1961. 7½×5. 2,732 pp. Price 87s 6d.

The annual revision of Kempe's Engineers Year Book has been completed, and the 1961 version contains new information in most of its 84 sections. This book is so widely known as a comprehensive and valuable work of reference, that it will be necessary here only to detail the changes. A chapter entitled Motor Vehicles has replaced that which previously appeared under the heading Road Vehicles: it is completely re-written and enlarged. Another section that has been re-written is that under the heading Heating, in the chapter on Heating, Ventilation and Air Conditioning. Among the chapters that have been extensively revised are those on Heating, Optics, etc; Hydraulics (Mechanics and Fluids); and Units of Measurement. The additions to the last-mentioned chapter include a full range of equivalents and conversion factors for metric and Imperial units.

Noteworthy additions to other sections of the Year Book are references to purifiers and filters for liquids and air, developments in bearing materials—ptfe and MoS_2—arc-air cutting, stud welding guns, electrolytic grinding, and metal deposition. In other sections there are additions on wood preservatives and treatment, road construction, foundations, surfacings, water storage and yield from reservoirs, compensation water, and design data for concrete dams. In the railway engineering sections there are additions on rails and rail fastenings, and in other sections, gas distribution machinery, blowers, compressors etc, coal ploughs and conveyors, power loading machines, ships contracts, universal beams, universal columns, sintered metal clutch facings, and sprag clutches.

Production of Wide Steel Strip

London: THE IRON AND STEEL INSTITUTE, 4 Grosvenor Gardens, S.W.1. 1960. 11×8¾. 187 pp. Price 65s.

A book entitled "Production of Wide Steel Strip" has recently been introduced by The Iron and Steel Institute. It is the report of a symposium held in London from 3rd to 5th May 1960. Wide strip mills are, of course, essential for the production of steel for, among other things, motor vehicles. A great deal of interest is being expressed, therefore, in the new steel making processes and in continuous casting, which render it possible to produce slabs for wide strip at lower capital cost than in the past with more conventional equipment. A valuable aspect of the book is that it includes the discussions, which represent a useful exchange of information between members who attended the symposium.

The contents are as follows: Future requirements of the tinplate industry; Future requirements of the motor-car body industry; Oxygen steel-making for flat-rolled products; Open-hearth steel-making practice for flat-rolled products; Production of steel with intensive use of oxygen in the open-hearth melting shop at Cornigliano; Quality control of wide strip mill products; Layout and design of a modern hot strip mill; Operations of the wide strip rolling mills at Fairless Works; Metallurgical principles and practice in the manufacture of cold-reduced tinplate; The rolling of very thin tinplate; Some engineering and operating problems in the conversion of a tandem cold-reduction mill to higher speed; Short-time annealing characteristics of capped, rimmed, and semi-killed wide strip; Annealing of cold-reduced sheet and coil at the Steel Company of Wales Ltd; The application of tin coatings in the manufacture of tinplate; Galvanizing: a review of the hot-dip processes with some reference to electrogalvanizing; and Plastic-coated steel: a promising new sheet material.

Drawing Office Organization

London: BRITISH INSTITUTE OF MANAGEMENT, Management House, 80 Fetter Lane, E.C.4. 1960. 9½×6. 55 pp. Price 15s. (12s. to BIM members and collective subscribers).

Originally, British Standard 1100: Part 9 was taken over and published as "Drawing Office Organization" in the BIM Production Management series of booklets. This was in 1951, and since then there have been various developments that tended to make the earlier work out of date. The new edition therefore has been prepared as a complete revision of its predecessor. In the introduction, it is stated that this publication has been designed primarily to assist managements

of engineering companies in making decisions regarding the organization, operation and control of drawing office activities. Architectural and cartographical drawing offices are considered to be outside its scope, although many of the recommendations and comments will, of course, apply to all types.

As can be seen from the list of contents, the work is comprehensive in its scope. The chapter headings are: Premises; Lighting; Layout of drawing offices; Drawing materials; Staff organization; Drawing practice; Drawing office routine; Planning and control of drawing office work; Standards and standardization; Reproduction of drawings; and Print room. Furthermore, there are two Appendices: the first deals with printed forms and the second is a bibliography and sources of further assistance.

The Mechanics of Vibration

By R. E. D. Bishop and D. C. Johnson

London: CAMBRIDGE UNIVERSITY PRESS, Bentley House, 200 Euston Road, N.W.1. 1960. 10×7½. 592 pp. 120s.

The primary aim of this book is to give the reader an insight into the vibration characteristics of a machine or structure rather than to present him with a routine for formal calculations. In developing the theory, first the concept of steady forced motion is dealt with and then free motion is introduced as a limiting case. Thus the classical theory of small oscillations is covered in a new way.

The concept of receptance, or dynamic flexibility, is used throughout and provides a link between the treatment of very simple systems, multi-freedom systems with lumped parameters and infinite-freedom systems. It also provides a tool for breaking down complex problems into simpler parts.

Students taking an honours course in engineering will be able to understand the mathematics, and the book should provide them with a background for their studies and with a path along which to advance for further work. Because the purpose of the book is to explain the means by which real problems may be tackled, rather than to give standard methods for answering particular questions, it should also be of value to practising engineers, post graduate students and teachers.

The list of contents is as follows: Introduction; Generalized co-ordinates and Lagrange's equations; Systems having any finite number of degrees of freedom; The taut string; The analysis of real systems—approximate methods; Torsional and longitudinal vibration of uniform shafts and bars; Flexural vibration of uniform beams; Viscous damping; Hysteretic damping; Free vibration; Non-harmonic and transient vibration; and Index.

Basic Synchros and Servomechanisms, Parts 1 and 2

London: THE TECHNICAL PRESS LTD, 112 Westbourne Grove, W.2. 1960. 9½×6. 118 and 111 pp. Price 16s. each part.

In July 1960 we published a review of Basic Electricity. The two volumes now under review are in general similar in form to this earlier work. The aim is at illustrating in the simplest possible way the fundamental characteristics of synchros and servomechanisms, used not only in many types of military equipment but now coming into operation in connection with automatic process control.

The purpose of the manuals is limited to the training of technicians for operating and repairing these mechanisms. To this end, they present a simplification of an otherwise complex set of subjects. Complicated mathematics has been avoided, and one of the primary aims has been exceptional clarity of text and illustration.

The contents of Part 1 are: Introduction to synchros; Synchro torque transmitters and indicators; The synchro differential; The synchro capacitor; Review of synchro fundamentals; Introduction to servomechanisms; An elementary servo; Redesign—with balanced potentiometers; Redesign again—with synchros; A basic transducer positioning servo; The component systems in a basic servo; and Index.

Part 2 contains the following sections: The function of the error-detector; The balanced potentiometer error-detector; The control transformer error-detector; Review of the error-detecting system; The function of the controller; The d.c. servo motor reviewed; The d.c. servo amplifier; Gas-filled valves in servo amplifiers; A thyratron motor control circuit; A thyratron servo amplifier with amplitude control; Phase-shift control in servo amplifiers; A thyratron servo amplifier with phase-shift control; The Ward-Leonard control system; The amplidyne; The a.c. servo motor; A basic a.c. servo amplifier; An a.c. servo amplifier with thyratron valves; An a.c. servo amplifier with vacuum valves; Oscillations in servo systems; Inaccuracies of synchro output; General review of servo fundamentals; Introduction to basic radar; and Cumulative index.

Manipulation of Thermoplastic Sheet, Rod and Tube

By J. M. J. Estevez, B.Sc, F.R.I.C, A.P.I, and D.C. Powell, B.Sc.

London: ILIFFE BOOKS LTD, Dorset House, Stamford Street S.E.1. 1960. 8¼×5½. 148 pp. Price 27s. 6d.

This volume is the third of a series of monographs which, although planned primarily to cover the syllabus in the technology, chemistry, physics and engineering of plastics leading to the Diploma, Graduateship and Associateship of the Plastics Institute, will also be found invaluable to anyone wishing to obtain specialized information on a particular subject connected with plastics. Each of the monographs is written by one or more experts in their own subjects, who are Members of the Plastics Institute; the chemical monographs are under the general editorship of Professor P. D. Ritchie, and the engineering ones of Mr. H. Barker.

The book is a clear and eminently readable survey of the principal thermoplastic materials available commercially, and the methods by which they can be fabricated into useful goods. Wherever possible, a critical assessment is made of the materials and techniques available for a particular application. The substances considered are polymethyl, methacrylate; polyvinyl chloride and vinyl copolymers; polystyrene and toughened polystyrene; and polythene. A detailed exposition of their properties—both of the virgin material and the end product—is followed by sections on machining, decoration and bonding.

Then the authors pass on to the important subject of forming, the materials discussed being divided into two classes: those suitable for oven techniques and those for vacuum forming. Further chapters deal with stamping and simple shaping aided by heat, the use of air pressure and vacuum, mechanically applied pressure, and the more complex processes involving a plurality of methods. Separate sections are devoted to the revolutionary development in shaping, the vacuum forming machine, and to tool construction and design. The last chapter deals with special topics such as the use of acrylics in the aircraft industry. Finally, the authors look to the future and conclude that, however strong the competition from injection and extrusion techniques, the unique feature of the shaping process—low tool costs permitting economic production in small quantities—assures it a permanent place in the plastics industry.

After the Preface and Introduction, the contents are as follows: Machining of sheet, rod and tube; Decoration of sheet, rod and tube; Joining by cementing and welding; The theory of heat forming thermoplastic materials; Simple shaping aided by heat—the use of air pressure and vacuum; The use of mechanically applied pressure; Complex processes using a plurality of methods; The vacuum forming machine; Tool construction and design; Special topics; and Index.

Air-Hydraulic Brake Servo

Compact Unit Comprising a Compressed Air and Hydraulic Servo and Air Reservoir

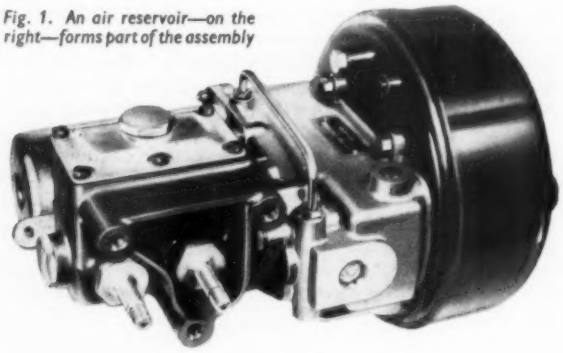


Fig. 1. An air reservoir—on the right—forms part of the assembly

A COMBINED compressed air and hydraulic servo unit, Fig. 1, for power assisted braking has been introduced by Clayton Dewandre Co. Ltd, of Lincoln. Since the brake pedal linkage is connected by a pull-rod to the lever assembly of the servo, the manufacturers term this a pull-type servo unit. Compressed air is used to supplement the effort applied by the driver, through the medium of the pedal, to the master cylinder to produce the hydraulic pressure for actuating the brakes. Thus, under all normal operating conditions, the hydraulic output pressure is proportional to the total efforts applied by the driver and the air servo. However, should the pressure fail, the brakes can still be operated hydraulically under the influence of the driver's effort alone. This pull-type air-hydraulic servo is interchangeable with an equivalent pull-type vacuum-hydraulic unit.

Illustrated diagrammatically in Fig. 2 is the installation of the servo unit and an engine-driven, S.C.4 air compressor. The output from the compressor is delivered to an air reservoir, which is part of the servo unit assembly. A feed-back pipeline is taken from the reservoir to the unloader mechanism built into the cylinder head of the compressor. There is also a safety valve for the system and it, too, is mounted on the compressor head. To warn the driver in the event of the air pressure's falling, a pressure-actuated electrical switch is installed on the reservoir but, of course, is not supplied by Clayton Dewandre.

The hydraulic master cylinder is of the tandem type and consists of a single cylinder containing two pistons. Each piston actuates a tip valve, through which oil is drawn from an independent compartment in the hydraulic reservoir, for operation of the brakes of one axle only. Thus, the brake systems serving each axle are independent in that loss of fluid, as a result of a breakage or leak, from the system serving one axle will not affect the operation of the other.

As in most systems, the degree of application of the brakes is proportional to the effort applied to the pedal, but the travel is influenced by the state of adjustment of the brakes. The servo assistance is, of course, proportional to the effort applied by the driver to the pedal.

General description

Basically, the assembly comprises four main units. These are the lever housing, on one face of which is mounted the air pressure reservoir, and on the opposite face the other two units, which are the hydraulic master cylinder and the air servo cylinder, with their axes horizontal and parallel.

The air cylinder is a flanged cup-shape pressing, the nominal inside diameter of which is 3·75 in. Its open end fits over a spigot on the lever housing. Around this spigot is a groove for an O-ring to form a seal at this end. Inside the cylinder are the piston and its return spring. The piston is an assembly of steel pressings, the arrangement of which can be seen from Fig. 3. Around it is a felt lubricator ring and a U-section rubber seal, with a flanged steel ring separating them. At the centre is connected the piston rod, around which is an extension sleeve, which is slightly longer than the stroke of the piston. This sleeve projects through a bush into the lever housing, where the other end of the piston rod is connected to the end of the reaction lever assembly. A U-section rubber seal, adjacent to the bush, prevents leakage of air along the periphery of the sleeve. The space on the pressure side of the piston is connected by what is termed the control tube to the control valve, and the communication between the other side of the piston and atmosphere is through holes in the centre of the piston, the bore of the extension sleeve, to the lever housing, and thence through a drain hole in which is fitted a split pin, which can be seen in Fig. 4.

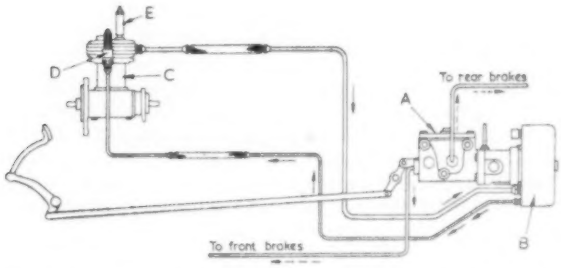
Levers and control valve

The main housing contains, in addition to the levers, a control valve. Basically, the lever assembly, Fig. 4, comprises two parallel pairs of arms, one pair being between the other two arms. Of these, the inner pair is termed the reaction lever assembly, and the outer pair the brake lever assembly. Approximately midway between the ends of the arms comprising the reaction lever assembly is a pin, which carries a spherical joint to connect them to the brake pull-rod, which is situated between the air and hydraulic cylinders. The reaction lever assembly is also coupled by means of a pin, at its end nearest to the servo piston rod, to the brake lever assembly. Another pin, immediately outboard of this, connects the whole lever assembly to the piston rod. At the opposite end, the brake lever assembly pivots on a support pin carried in the housing. Around the central portion of this pin, but clear of it, is a sleeve that carries a roller between the ends of the inner pair of levers, which it also interconnects. This roller actuates the control valve. The other pin connection, carried in bushes in the outer pair of arms, and passing through clearance holes in the inner pair, is that to the pushrod of the hydraulic cylinder.

The control valve can be seen in Fig. 3; it comprises two main components: one is a spring-loaded plunger, which is actuated by the roller previously mentioned, and which forms the atmosphere valve; the second is the inlet valve, which is of the mushroom type and has an axial hole through it. These two valves are in line in a common housing. On the atmosphere valve, the end remote from that against which the roller bears is rubber faced where, when the valve is depressed, it contacts the end of the stem of the inlet valve, thus sealing the axial hole through it. Compressed air is piped to a radial duct leading to an annular space machined around the stem of the inlet valve. It is prevented from escaping along the stem towards the atmosphere valve by an O-ring in a groove around the stem.

The other end is flanged, to form the head, and carries a rubber seating. When the inlet valve is seated, the pressure acting in both directions — that is, towards the seat and towards the end of the stem—holds the valve in a balanced condition, so the crack-off pressure of the valve is low. The port to atmosphere, which is in the side of the lever housing and which is ducted to the space between the atmosphere valve and the inlet valve, is fitted with a wire mesh filter.

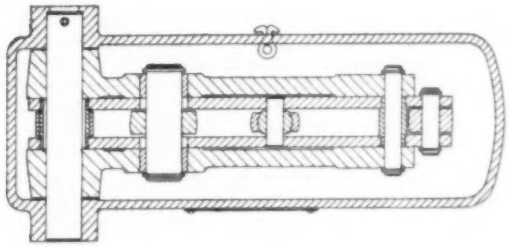
As can be seen from Fig. 3, when the brakes are released the air flows back from the servo cylinder, through the external pipe connection, into the chamber containing the inlet valve return spring, through the axial hole in the inlet valve to the space between it and the atmosphere valve and thence, out through the wire mesh filter. When the plunger that forms the atmosphere valve is depressed by the roller it comes up against the end of the inlet valve, closing the axial hole and lifting the valve off its seat. This allows compressed air to pass from the annular space around the inlet valve into the chamber in which its return spring is housed, and thence through the pipe connection to the servo cylinder, which can be seen in the illustration opposite.



A *tandem hydraulic master cylinder;* **B** *air reservoir;* **C** *S.C.4. compressor;* **D** *governor* **E** *safety valve*

Fig. 2. Above: A typical braking system layout for use with the pull type servo and the Clayton Dewandre S.C.4 engine-driven compressor

Fig. 4. Below: Illustration showing details of the lever assembly



Air reservoir

For protection against corrosion, the air reservoir is stove-enamelled both inside and out. It is furnished with tapped bosses for the connections for the compressed air supply, the governor feed-back pipe and the reservoir drain plug; another tapped boss carries the low-pressure warning switch. A special feature of the drain plug is that air and condensate can be blown-off without completely removing the plug. Four studs are projection welded to the reservoir, for attaching it to the lever housing.

Hydraulic master cylinder

Illustrated in Fig. 5 is the hydraulic master cylinder. As can be seen from Fig. 1, it incorporates the mounting bracket for the complete servo unit assembly. In the cylinder, the bore of which is 1·75 in, there are two pistons, one of which transmits the hydraulic pressure to the front and the other to the rear brake cylinders. Above and integral with the cylinder is the fluid reservoir which is divided in two by a partition, each portion serving one part of the hydraulic system. The two recuperator valves, one in each section of the reservoir, are of the tip type and are screwed into the base so that their spindles project into the cylinder beneath. When the brakes are applied, the flange of each piston moves away from the tip valve spindle adjacent to it, thus allowing the valves to close. This traps the fluid in the cylinder and, as the pressure builds up, it applies the brakes. When the pistons return to their off positions they again tip the valves and thus open them, so that compensation can be made for any loss of fluid, or for thermal expansion or contraction. Each piston is fitted with two U-section seals. On the primary piston, the lips of both of the seals are turned towards the pressure chamber, whereas on the secondary piston the seals are fitted back to back: the reversed seal on the latter piston is necessary, of course, to prevent the passage of fluid past this piston in the event of leakage from the secondary hydraulic system. A return spring bears against one end of each piston. The outlet from the primary cylinder is a long-reach connection in the mounting bracket, whereas, that from the secondary cylinder can be fitted either at the end of that cylinder, as in Fig. 5, or beside the mounting bracket, as in Fig. 1. In each con-

Fig. 3. Elevation and longitudinal section of the pull type air-hydraulic servo. The atmosphere and air inlet valves are in the lever housing

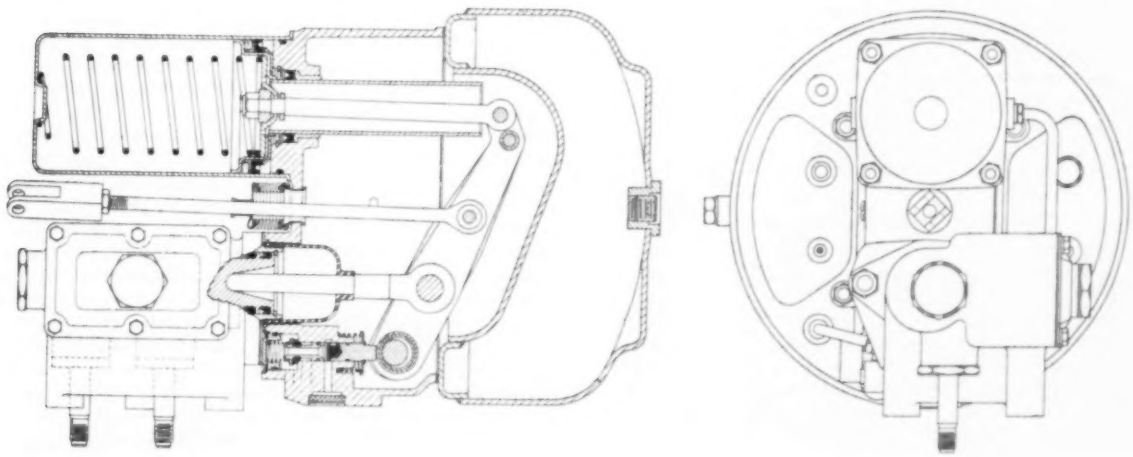


Fig. 5. Right: The hydraulic master cylinder is of a new design by Clayton Dewandre. It is a compact tandem type unit in which the front and rear brake systems are served by separate compartments in the reservoir, so that in the event of failure causing loss of hydraulic fluid from one portion of the system, the other portion is unaffected

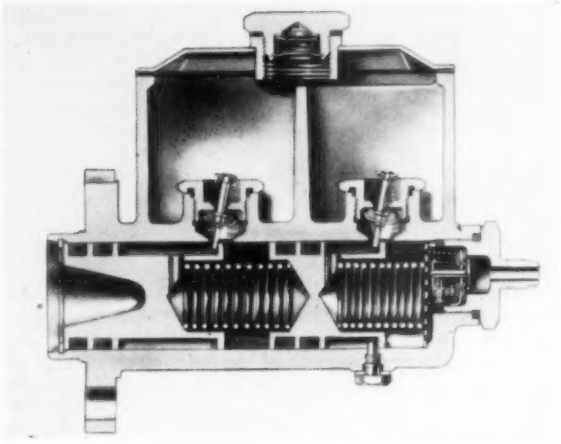


Fig. 6. Below: Diagrammatic illustration showing the condition of the servo when the brakes are released and neither the direct nor the servo efforts are being applied to the hydraulic master cylinder

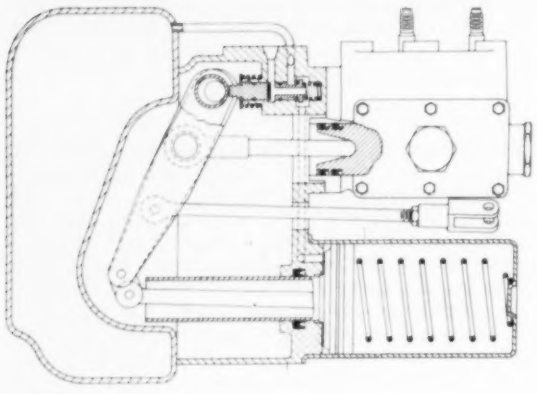


Fig. 7. Below: The initial condition when the brakes are applied

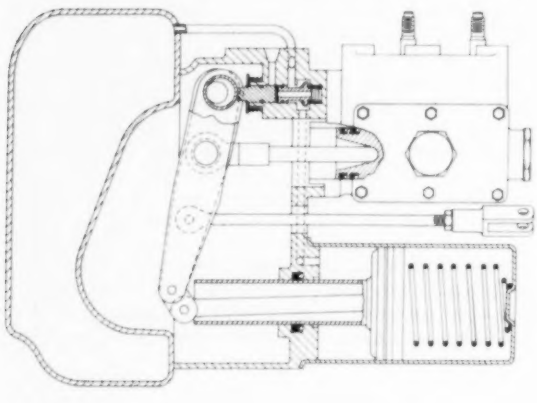
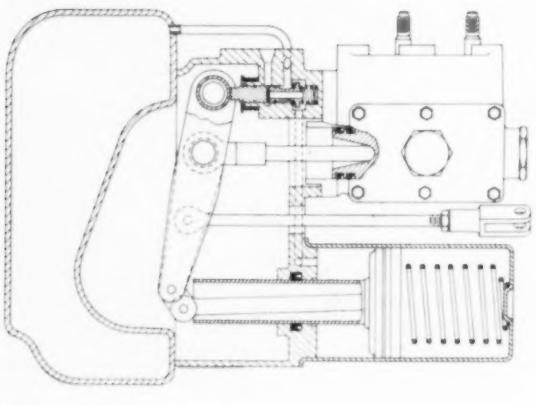


Fig. 8. Below: This diagram illustrates the balanced condition



nection there is a residual pressure valve, which maintains a slight pressure in the brake lines to prevent air from entering the system. This arrangement also assists in the bleeding of the hydraulic system since fluid pumped from the master cylinder cannot return, and this ensures that a fresh charge of fluid is delivered with each stroke of the pedal.

Clearance between the pushrod and its seating in the hydraulic piston is regulated by shims between the joint faces of the master cylinder and the lever housing. A circlip in one end of the cylinder forms the return stop for the primary piston, and a return stop for the secondary piston is screwed radially inwards through a hole in the cylinder wall. The stop to limit the forward motion of the secondary piston is a perforated plate at the other end of the cylinder, and that for the primary piston is the rear end face of the secondary piston. In the cover on top of the reservoir there is a boss for the filler cap with its non-return valve, which admits air to the reservoir while preventing loss of fluid owing to splashing.

Operation

When the brakes are released, the power piston moves under the influence of its return spring until it is stopped by the end wall of the lever housing, and simultaneously the levers move towards the opposite wall of the housing, Fig. 6, that is, towards the reservoir, until the roller is moved clear of the end of the control valve; thus the compressed air inlet valve is seated and the atmosphere valve open. Under these conditions, both sides of the power piston are at atmospheric pressure and no effort is being applied to the hydraulic master cylinder, in which both pistons are therefore held against their stops by their return springs.

Initially, when the brakes are applied, Fig. 7, the end of the reaction lever assembly connected to the power piston remains stationary, since it is held at the extreme end of its travel by the piston return spring. Therefore, depression of the brake pedal and the consequent withdrawal of the pull-rod causes the reaction lever assembly to pivot about the pin that connects it to the power piston rod. At the same time, the roller at the end of the lever assembly moves through the clearance and seats the atmosphere valve on the port in the end of the inlet valve. This isolates the pressure side of the power piston from atmosphere. Further movement unseats the inlet valve so that compressed air from the reservoir is admitted and passes through the control tube to the air cylinder. The point at which the inlet valve lifts from its seat is called the valve crack-off point.

The pressure differential, on the two sides of the power

piston, moves it along the cylinder, and pulls the levers to augment the effort applied through the pushrod to the hydraulic cylinder. Initially, the primary piston is driven along the hydraulic cylinder, closing the tip valve and forcing fluid to the rear wheel brake cylinders, until the shoes contact the drums. Additional effort applied to the primary piston increases the hydraulic pressure and moves the secondary piston, closing its tip valve and forcing fluid into the front wheel brake cylinders until their shoes also contact the drums. Further effort generates equal pressures in both the primary and secondary cylinders so that the load to all brakes is fully compensated.

During the brake application, a point is reached at which the pressure differential across the power piston, transmitted through the piston rod, balances the effort on the pull-rod. The ratio of the two forces required to effect the balance is, of course, determined by the leverage about the pin that connects the reaction and brake levers. In this balanced condition, the roller at the other end of the reaction lever assembly is centralized relative to the lever support pin, so the clearances are equalized and the inlet valve is closed, Fig. 8. If additional force is applied to the pedal, the unbalanced condition occurs again and the valve opens until the balance is once more restored.

Since the reaction force is transmitted back to the brake pedal, the driver is able to sense the degree of application. Moreover, as the pistons move along the hydraulic master cylinder, the lever assembly, and with it the pedal, moves in the same direction, so the driver also has a sense of the position of the brake control while he is applying it.

FIRTH CLEVELAND FAN

A NEW design of fan, which appears to have considerable merit, has recently been introduced by the Firth Cleveland Group. It is known as the Tangential Fan and is based on German patents, for which Firth Cleveland has acquired the exclusive rights in the United Kingdom. Though intended primarily for moving air, the fan is also suitable for other gases, vapours and fluids. It will be manufactured by The Solartron Electronic Group Ltd, of Farnborough, Hants, a member company of the Firth Cleveland Group.

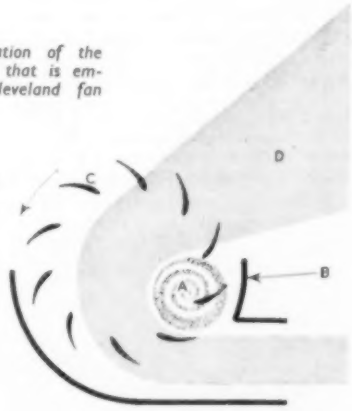
The two main disadvantages of the conventional type of fan having a low-speed delivery characteristic are a relatively high noise level and a low efficiency; both are due to the stalled condition of the air round the rotor blades. In the Tangential Fan they are overcome by the use of what is called the tangential flow principle. An external guide vane causes a stationary vortex to be induced eccentrically within the rotor of the fan: the axis of the vortex is parallel to that of the rotor, and it is a little nearer that axis than is the inner circumference of the rotor blades; the envelope of the vortex encroaches on the volume swept by the blades. Since the peripheral speed of the vortex is greater than that of the blades, the air is accelerated as it enters the rotor, thus avoiding any stalling of the stream. The vortex then imparts an angular displacement to the air stream before ejecting it tangentially in a virtually laminar flow, which has considerably greater carrying power than has the turbulent flow that is delivered by a fan of the orthodox type.

In relation to its output, the fan is unusually compact because of the small diameter of the rotor, which is of the squirrel cage type and is a plastics moulding. The vortex-forming guide vane and the outlet diffuser are of sheet metal. A shaded-pole type electric motor is available to drive the fan. The makers estimate that, for a given power input, the air delivery should be at least double that of a fan of traditional design; the pressure head should be 2 to 2½ times as great and the noise reduced by about 90 per cent.

Initially, manufacture will be concentrated on domestic appliances, such as room heaters and hair driers, in which the fan is used in conjunction with a heat exchanger of unusual design. However, many other applications, both automobile and industrial, are scheduled for subsequent development. Among these are car heating and air-conditioning equipment, fluid couplings, and the cooling of air cooled engines. It is considered by the manufacturers that the freedom from churning of the fluid will be highly advantageous in a fluid coupling, and the efficiency of the fan for engine cooling purposes has already been demonstrated by a series of tests carried out in Germany on a 63 b.h.p. engine: the standard cooling fan absorbed 6·8 b.h.p. at the peak speed, whereas the tangential flow unit required only 1·8 b.h.p—a saving of 5 b.h.p. or almost 8 per cent of the output.

Because the heat exchanger previously mentioned is likely to be employed in industrial and perhaps automobile applications, it merits a brief description here. The electrically heated element operates in the black-heat range of temperature, by virtue of its large surface area and minimal obstruction to the air stream. To obtain this area, a metal ribbon is edge-wound into a helix; instead of being flat, the ribbon has a longitudinal joggle, which increases the stiffness of the helix sufficiently to render a core unnecessary. The element is fitted in the fan diffuser, with the helix axis perpendicular to the direction of the flow of the air.

Diagrammatic representation of the tangential flow principle that is employed in the Firth Cleveland fan



A *induced vortex;* **B** *vortex-forming guide vane;* **C** *blades of fan rotor;* **D** *incoming air stream*

Bonded Body Panels

BONDED construction has recently been employed by Bonallack Ltd, for the side and rear panels of a truck body. It obviates the unsightly cracking of paint around rivet or bolt heads; also, by virtue of the use of this form of construction and of piano hinges, the sides of the body are completely free of projections, so that both painting and the subsequent cleaning are simplified. The panels are of aluminium and they are bonded to internal ribs with Redux adhesive, made by CIBA (A.R.L.) Ltd., of Duxford, Cambs.

New Plant and Tools

Recent Interesting Developments in Production Equipment

A RANGE of machines for making coils, bends and circles in bars, tubes, and various sections has recently been introduced by Funditor Ltd, Woodbridge House, Woodbridge Street, London, E.C.1. This range is divided into the two classifications of coilers and benders. In the range of coilers there are machines of nine sizes; they are available in either vertical spindle form, designated model C, or horizontal spindle form, designated model DC, and there is a variable speed drive on each. Not only coils, but also U-shapes, spirals or compound bends can be made.

One of the chief features of these machines is that, for a given size of stock, only one set of rollers is required, regardless of the radius of bending. The illustration is of a type C 145 coiler, which, among other things, can accommodate solid mild steel bar of $\frac{1}{2}$ in diameter, copper tubing of 1 in diameter and steel channel, with flanges uppermost, of 1 in × $\frac{1}{2}$ in × $\frac{1}{4}$ in. Mild steel bar in sizes from $\frac{7}{8}$ in diameter to $2\frac{1}{8}$ in diameter is dealt with by the other machines in the range, with correspondingly different sizes in other sections and metals. Forming rollers must, of course, be individually designed to each shape of bar or section.

In the case of the C 145 model, power is supplied by a $1\frac{1}{2}$ h.p. motor, which drives one of three forming rollers. Stock is fed between this and another roller until it meets the third one, which is mounted on a radius arm that swivels about the axis of the driven roller. The radius of bend is dependent on the angular position of the arm, and the stock is formed into a circle unless another roller or pitch plate is placed in its path to deflect it upwards and thus create a continuous coil. It is a simple matter to swing the arm back to its rest position or an intermediate position, and thus interrupt the bend; similarly, the coil can be swung over during a run, in a manner such that subsequent bends are in a different plane.

Movements of the radius arms are effected pneumatically in the small models and hydraulically in the larger ones, and a manually adjustable stop is incorporated. At the front of each machine is a control panel, with push buttons to start a coil, run a straight length, curve it, and stop it at any point for inspection or change of radius. The machine is reversible, so that bends can be formed in both directions.

There are two types of bending machines: simple benders, designated B type; and double acting benders, designated DAB. They cover a range of sizes similar to those of the coilers, and can be either hand or foot controlled. The production rate of B type benders, in standard form, is from 300 to 1,000 bends per hour, but this figure can be considerably improved if several pieces are bent at once.

As with each of the coilers, a manually adjustable stop is incorporated; however, in the case of the bending machines there is available, as an alternative fitting, a four-place turret stop which permits four different bends to be made in rapid succession by the same fixture.

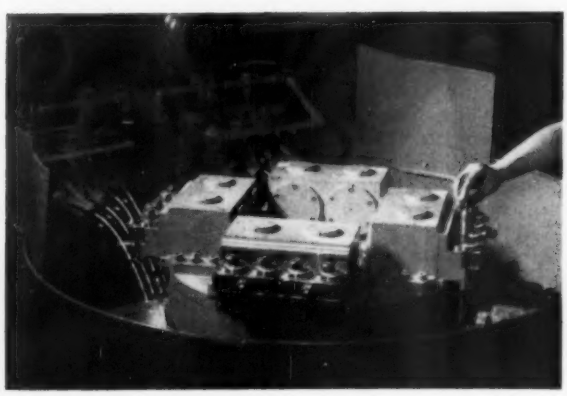
Type C 145 Coiling machine, showing the tilted pitch plate which deflects the stock upwards as it leaves the rollers, thus forming a rising coil. Only one set of rollers is needed for a given stock section, regardless of the radius of curvature and the pitch of the coil that are required
(Funditor Ltd.)

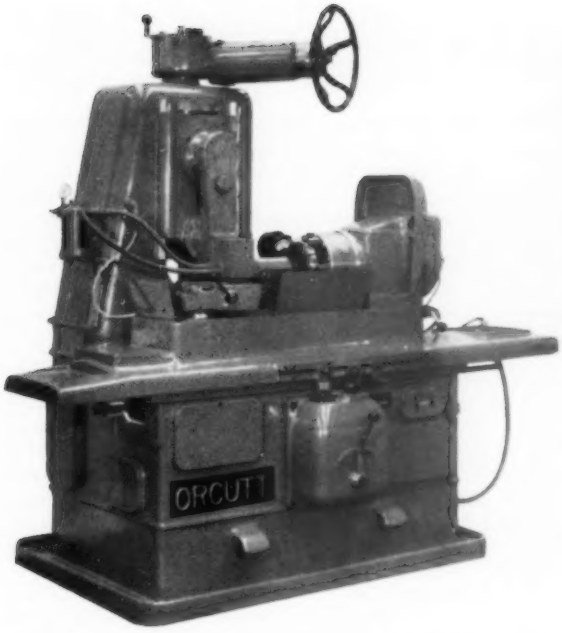
Special-purpose, multi-spindle drilling machine

At one of the factories of Leyland Motors Ltd, Leyland, Lancs, a new machine has been added to the plant that is engaged in the manufacture of engine components. It is a four-station, continuous cycle unit for drilling split-pin holes in connecting rod bolts, and it has been designed and built entirely by Leyland engineers to supersede the two machines previously used for this purpose. With this new machine, output is increased, operator fatigue is less than it previously was and the life of drill bits is much longer.

A four-sided indexing turret is the central feature of the machine. On each of its sides there is a jig containing four vertical bushed holes into which the bolts are dropped, heads uppermost, to be held by pneumatically operated clamps. These bushes are interchangeable, and available in various

Drilling split pin holes in connecting rod bolts for diesel engines
(Leyland Motors Ltd.)





Above: Close-up of the worktable. In the centre is the trimming device, and at the lower left-hand corner is a typical finished component

Left: The "Orcutt" spline shaft grinder modified to form internal longitudinal grooves. An L-shape internal grinding attachment has been fitted to the head to replace the quill that is standard equipment
(Gear Grinding Co. Ltd.)

sizes to accommodate all the connecting rod bolts for the diesel engines in the Leyland range. The holes to be drilled are, of course, at the ends of the bolts, and each bolt has two holes that intersect at right angles.

At the beginning of the work cycle, four bolts are placed in a jig at the loading station, and clamped; the turret is indexed round until it faces the first of two four-spindle drilling heads; then the drills advance, and one hole is drilled in each bolt. As the jig is indexed round to face the second drill head, the next jig on the turret takes up its position facing the first drill head. During drilling operations the jigs do not face the drill heads squarely, but are at 45 deg, as viewed in plan. Therefore, since the eight drills are parallel, the second drilled hole in each bolt is at 90 deg to the first. At the end of the cycle, the jig arrives at the loading station, the clamps release the bolts, and four pneumatic plungers underneath the jig raise the bolts clear of their holes to enable the operator to remove them.

Internal groove grinding

One of the machining processes in the manufacture of the Birfield constant velocity universal joint is the grinding of internal longitudinal ball tracks. A standard "Orcutt" 20 in spline shaft grinding machine has been adapted for this purpose by its makers, the Gear Grinding Co, Cranmore Boulevard, Shirley, Warwickshire. An internal grinding wheel, mounted at the end of an arm suitably shaped to enter the bore of the workpiece, has been fitted in place of the standard spindle quill, and a profile trimmer now replaces the original spline trimmer.

The four longitudinal tracks to be ground are in a $3\frac{5}{8}$ in diameter bore and are of part-elliptical section. The generation of this elliptical profile on the grinding wheel is relatively simple, for the trimmer is set at an angle to the axis of the wheel and describes an arc. Hence, the width of the wheel corresponds to the minor dimension of the ellipse. In order to reduce setting-up time to a minimum, a diaphragm chuck is employed to hold the workpiece, and it is embodied in an extremely accurate indexing assembly at one end of the worktable. Compressed air at 75 lb/in^2 is used to open the chuck jaws, but diaphragm pressure keeps the workpiece firmly clamped throughout the grinding operation. After each stroke of the grinding wheel, the component is indexed through 90 deg, and the stroke is repeated. At each complete revolution the feed depth is increased by 0·0015 in and four more strokes are made, until the full depth in all four tracks is attained.

Lubrication for the spindle and layshaft bearing assemblies is by a fine mist spray of oil forced through an atomizer by compressed air at 5 lb/in^2; subsequent to being sprayed, the oil is piped into the wheel head. As in the standard machine, a 3 h.p. motor drives the grinding wheel, which revolves at 6,450 r.p.m, to give a cutting speed of 5,500 ft/min.

Guillotine shears

To satisfy the demand for guillotine shearing machines suitable for application in rapid production systems, Rushworth & Co. (Sowerby Bridge) Ltd, of Sowerby Bridge, Yorkshire, have introduced their "Series 25" range, which can accommodate mild steel plate up to $\frac{1}{4}$ in thick in widths from 6 ft to 12 ft. Accurate setting of blade clearance is afforded by adjustment of the block on which the lower blade is mounted, and since the slideways of the moving

Shearing machine which accommodates $\frac{1}{4}$ in plate up to 12 ft wide
(Rushworth & Co. (Sowerby Bridge) Ltd.)

(Soag Machine Tools Ltd.)

This portable bandsaw can cut through a 1½ in steel bar in 80 seconds

blade are inclined at an angle of 2 deg to the vertical, the movement is downwards and backwards and thus, it is claimed, the plate is cleanly sheared. The speed of operation of the machines is 60 strokes/min.

There are two cantilever arms at the front, for supporting the work, and a squaring arm can be provided if necessary. The standard hold-down stops are of the spring-loaded plunger type, but hydraulically operated plungers are available as an alternative. Another alternative item in the specification is a power-operated precision back gauge that is controlled by push buttons and equipped with a dial indicator to show the position of the gauge bar.

All plain bearings and wearing surfaces are lubricated by a one-shot system, and the driving worm and wormwheel run in an oil bath, so maintenance is reduced to a minimum. A foot-operated starting switch, at the end of a flexible armoured cable, is the standard control for the driving motor; this feature is particularly useful when long plates are being sheared, and the operator is not within arm's length of the machine. If the machine is to be used on fairly small work, this switch can be replaced by push buttons situated on the front of the machine or on a pendant control box. Lighting is incorporated behind the front guard, and a wire throws a shadow on the exact cutting line so that, when a large plate is being sheared, and the operator is not close enough to see the shearing area clearly, the work can still be performed with accuracy.

Magnetic plastics strip

As an addition to their range of permanent magnets, James Neill & Co. (Sheffield) Ltd, Napier Street, Sheffield 11, have introduced a new product, the "Eclipse" permanent magnet strip, which is of flexible plastics material impregnated with magnetic powder. The strip, which is of rectangular section, has a longitudinal groove in one face; this is the face with the strongest magnetic pull and, as shown in the illustration, it has transverse polarity.

This strip can be cut with a knife or drilled, and has wide application in workshops and on production lines. For example, small ferrous parts can be carried, without risk of slipping, up steep conveyors embodying short magnetic strips; masking shields can be held in place on vehicles receiving two-tone paint treatment; small components can be picked up more easily if industrial gloves incorporate magnetic material; engine sumps normally equipped with magnetic plugs can be fitted with a small piece of strip more cheaply. The manufacturers stress that the magnetism is permanent, and can be destroyed only by burning; if overheating does occur, it affects the plastics binder before the magnetic properties become permanently affected. Flexibility at room temperature is such that the strip can be bent to a 3 in diameter circle without breaking; this flexibility is increased by warming, and the strip, after being reduced to room temperature, retains the shape to which it was bent when warm.

The cross-sectional dimensions are $\frac{5}{16}$ in × $\frac{11}{64}$ in, and the material is marketed in flat cardboard dispensers, each containing 12 ft of strip. An approximate weight of 1·3 oz per ft is quoted and lifting capacity per inch of length is 3 oz when there is no gap between strip and workpiece, rising progressively to 1·5 oz when a 0·015 in layer of paint or plating is interposed. In applications such as that on the conveyor system, it is advisable to cement the strip to its backing material rather than to pin or nail it, as this obviates curling and bowing.

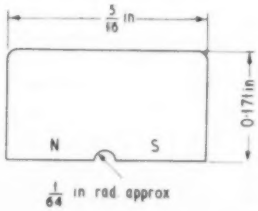
Portable bandsaw

A machine that may eventually replace the hacksaw in many industrial applications has been announced by Soag Machine Tools Ltd, Juxon Street, London, S.E.11. It is called the "Baby" portable bandsaw, and it can cut through a 1½ in diameter mild steel bar in 1 min 20 sec. Operation of the saw does not call for any exertion on the part of the operator; he merely rests the blade on the workpiece and lets the weight of the machine carry the cut through.

A built-in motor of ¼ h.p. drives the band at a cutting speed of 115 ft/min. Two pairs of hardened steel rollers guide the band, the tension of which is maintained by a rack and pinion system. When a band has to be changed, this tension can be instantly released by turning a button through 180 deg; it is claimed that changes should not be frequent, however, as bands have a long life and will not break in normal use.

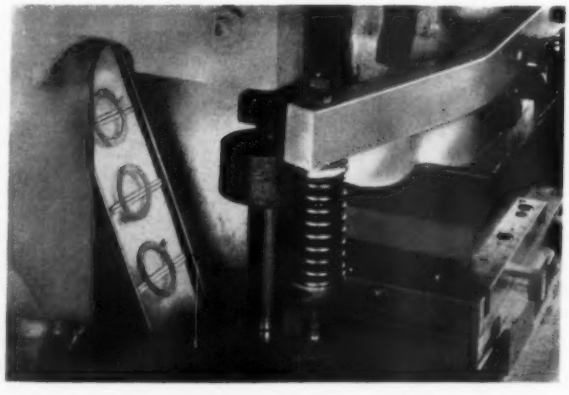
The machine, which is supplied complete with a saw band, weighs 21 lb, and its overall dimensions are 21 in × 7 in × 6¾ in. There is available, as an optional extra, a stand for converting it to a vertical bandsaw; alternatively, the machine can be turned into a horizontal bandsaw if this stand is mounted in a vice. Round bar of up to 2½ in diameter and sections of up to 4½ in × 3 in can be accommodated between the abutments of this machine.

Sectional view of plastics magnetic strip, showing direction of polarity over the grooved surface



Below: Magnetic strip attached to a steeply rising conveyor, and carrying small ferrous parts

(James Neill & Co.)



TRIUMPH HERALD 1200

Outstanding Feature of the Latest Series of Cars is an increase in the Swept Volume of the Engine

THE new Triumph Herald 1200 cars, introduced in April, supersede all the previous models except the low-price S saloon. Although for the present the range of body types available—saloon, coupé and convertible—is as hitherto, it is expected that a station wagon variant will go into production in the near future. The most important single change from the earlier series is that swept volume of the engine has been increased from 948 to 1,147 cm^3; this increase has been obtained by enlarging the bore from 63 to 69·3 mm without alteration to the stroke of 76 mm or the compression ratio of 8·0 : 1. Initially, only a single-carburettor version of the engine is being built, but a two-carburettor unit is under development.

In increasing the swept volume by 21 per cent, the manufacturers' main objective has been to effect a substantial improvement in the available torque over the normally used speed range. Greater peak power, which clearly only affects the maximum speed and the acceleration at high r.p.m, has wisely been regarded as of secondary importance. Consequently, the cylinder head and valve assembly and the manifolding of the smaller engine are retained. The increase in the maximum net torque is 27 per cent — 60·8 lb-ft as against 47·9 lb-ft—and the engine speed at which the torque peak occurs has fallen from 2,750 r.p.m. to 2,250 r.p.m. In contrast, the maximum net power output has gone up from 34½ to 39 b.h.p. at 4,500 r.p.m, a rise of only 13 per cent. The maximum net b.m.e.p. of the new engine is 131 lb/in^2, just under 5 per cent higher than the 125 lb/in^2 given by the smaller unit.

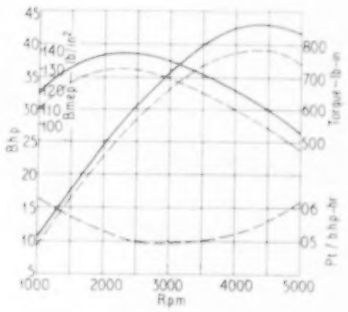
For economy in production, it was obviously desirable to avoid external alterations to the block casting, and for the existing transfer machining line to be used. When the bore was enlarged several years ago, from the 58 mm of the original Standard Eight engine, the cylinders of the front and rear pairs had to be siamesed, and it was not thought practicable to reduce the thickness of the consequent common walls. The only feasible method of obtaining a further increase of bore was, therefore, to move the axes of the end cylinders $\frac{5}{32}$ in further from the middle of the engine, and to bring those of the inner cylinders closer to it by the same amount.

However, the effect of this alteration was to bring the bores undesirably close to some of the cylinder head retaining studs. A virtue was therefore made of necessity, by shifting the cylinder axes $\frac{5}{32}$ in to the left of the crankshaft axis. This *désaxé* layout has, of course, the advantage of reducing any tendency to piston slap, by causing the piston to be transferred with less force from the non-thrust to the thrust side of the bore in the region of top dead centre at the beginning of the firing stroke.

As a result of the altered cylinder spacing, the crankshaft has been redesigned, to obviate the necessity of offsetting the big ends from the major axes of the connecting rods. By using copper-lead instead of white metal, it has been possible to reduce the nominal width of the big-end bearings from 1·09 in to 0·91 in, thereby avoiding any weakening of the webs adjacent to the main bearings. By virtue of the narrower bearings and the increased interval between the cylinders of each outer pair, the flying webs are now of more robust design, so the new shaft is, in fact, stronger overall than was its predecessor. The main bearings also are of the copper-lead type, but their dimensions are unchanged.

Because of this change of bearing material, the oil filter is now a full-flow unit, containing a paper element, whereas a by-pass filter was previously installed. In addition, the delivery pressure of the oil pump has been increased by 20 per cent at normal running speeds. Other engine modifications include the use of larger-diameter gudgeon pins than before, and the adoption of a Solex B30ZIC-5 carburettor, with a choke diameter of 30 mm, in place of the B28ZIC-2 instrument of the 948 cm^3 engine.

In view of the company's professed intention of providing a livelier car, it is perhaps surprising that a final-drive ratio as high as 4·11 : 1 has been adopted for all three; the old ratios were 4·875 : 1 on the saloon and 4·55 : 1 on the coupé and convertible, which had the two-carburettor power unit. At peak engine torque in top gear, the torque at the rear wheels of the new saloon is only about 7 per cent higher than it was on the earlier version. This comparison can hardly be extended to the coupé and convertible models of the two series, because the 948 cm^3 two-carburettor engine developed its maximum torque at a much higher speed, 4,200 r.p.m, than either the 948 cm^3 or the 1,147 cm^3 single-carburettor unit.



Performance curves for the 1,147 cm^3 Herald engine; the net values are indicated by the dotted curves. Whereas the torque is 27 per cent higher than that of the smaller engine, the increase in power output is 13 per cent

Although the top gear acceleration is perhaps restricted by the adoption of this high final-drive ratio, there is the valuable compensation of a substantial reduction in r.p.m. at any given cruising speed. As a result, the fuel consumption of the 1200 cars on the road is very little inferior to that of the lower-powered models. Also, acceleration through the gears is improved by the closer spacing of the ratios: formerly these were 1 : 1, 1·454 : 1, 2·46 : 1 and 4·271 : 1, whereas the new ones are 1 : 1, 1·397 : 1, 2·16 : 1 and 3·755 : 1.

Among the other changes made to the cars are improvements to the front seats and to the external and internal trim. Two commendable features, the Impactoscopic steering column and the adjustment of the driver's seat for height and rake, as well as longitudinally, are retained. Although the facia is still of pressed fibre, it now carries a panel of polished wood veneer, which gives the interior a more luxurious appearance. Finally, the integral bumpers are fitted with white covers of moulded rubber. These serve the dual purpose of enhancing the appearance and of concealing any indentations of the sheet metal caused by minor impacts. The original version of the Triumph Herald was described in the September and October 1960 issues of *Automobile Engineer*, on pages 352 to 362 and 392 to 403.

Machining the Cluster Gear for the B.M.C. ADO 15

4,500 Units per Week Produced by Churchill "Link-Line" Technique

IN July 1959, the British Motor Corporation installed a production line for the gear-cutting of the gearbox layshaft cluster gear of their then-new ADO 15 saloon. The line was a fully automatic one, built by Churchill Gear Machines Ltd, of Blaydon-on-Tyne, County Durham, in accordance with their "Link-Line" technique. It comprised eighteen machine tools served by conveying devices of the latest Churchill design.

A second "Link-Line", similar to, but incorporating improvements based on subsequent experience, is now to be installed at the B.M.C. factory. This line is planned to be 200 ft long and will consist of 23 machine tools. It is designed to produce, when operating at full capacity, 4,500 units per 70 hour week. A high degree of standardization has been aimed at, with the result that, in this production line, there are only two kinds of conveyor mechanism. Each of the 23 machines is started and stopped independently, either manually or automatically, according to the requirements in respect of work loading and tool replacement or maintenance.

The machines, in operational sequence, are: eight Churchill Mk IV Rigidhobber machines, five Fellows England type 7125 A gear shapers, three C.G.M. 8 in chamfering machines, two C.G.M. 8 in tooth-rounding machines, four Churchill type F.A.U. 8 in gear-shaving machines and one Newall type LA cylindrical-grinding machine. All the conveyors, gear washing and drying units, transfer arms and elevators, are manufactured by Churchill Gear Machines Ltd.

Since the hobbing and shaving machines have their work axes vertical, the blanks are conveyed standing on end. The approach to each of the subsequent machines, however, is a sloping ramp, to which the components are elevated, and down which they roll to the work area: in this section of the line, therefore, the axes of the components are horizontal, at 90 deg to the direction of travel of the conveyor.

Prior to delivery of components at the start of the first conveyor, which serves the hobbing machines, the peripheries of all gears have been skimmed and checked for diameter. Through the centre of each component is a hole which has its ends cone-centred for location in the work-holding centres during machining operations. Of the four gears on each cluster, three are helical and one is a spur gear; the helical gears are of 13·054 DP, and the spur gear is 10 DP.

This spur gear and two of the helical gears are cut on the hobbing machines, but the remaining helical gear, which is the second-speed gear, is of smaller diameter than the closely adjacent third-speed gear. A hobbing operation on this gear is impracticable, and five gear-shaping machines are therefore used instead. It is these machines, with their inherently longer cycle time, that determine the rate of production of the line.

When all four gears have been cut, the clusters move on to the three chamfering machines, where the ends of the helical teeth are deburred, and then to the two machines which round off the ends of the spur gear teeth. The teeth are finished on four shaving machines, one machine per gear, and these operations are followed by the final one, in which the peripheries of the three helical gears are ground, for accurate location during the subsequent grinding of the bore. The material of the cluster gears is En.351, ¾ per cent Ni Cr case-hardening steel, in this application.

Hobber and shaper conveyor

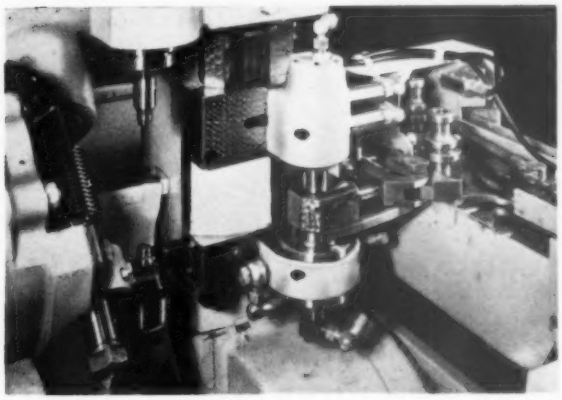
The "Link-Line" is in two sections, the first being served by two continuously moving conveyors, and the second by individual elevator-loader devices. Progress of the components is from right to left along the line of machines, as viewed from the side on which the conveyors are placed. Each of the two conveyors is divided by fixed partitions into three longitudinal channels, the right-hand channel being nearest the line of machines. The components fed on to the conveyor are carried in the central channel to a position opposite the first of the three hobbers that share the machining of fourth-speed gear, and there they are built up into a storage bank of 13, the maximum number the bank can hold. While the components are being held stationary in this way, the conveyor is, in effect, sliding beneath them.

Further components arriving at the bank are deflected, by the last component in it, into the left-hand channel and subsequently are diverted back into the central channel, to be fed into the storage bank of the second hobber, where the process is repeated; this latter bank holds 28 units. If the number of components in the storage bank of the third hobber rises to 53, the line is automatically stopped. In this way, the maximum number of components waiting to have fourth-speed gear machined is 94.

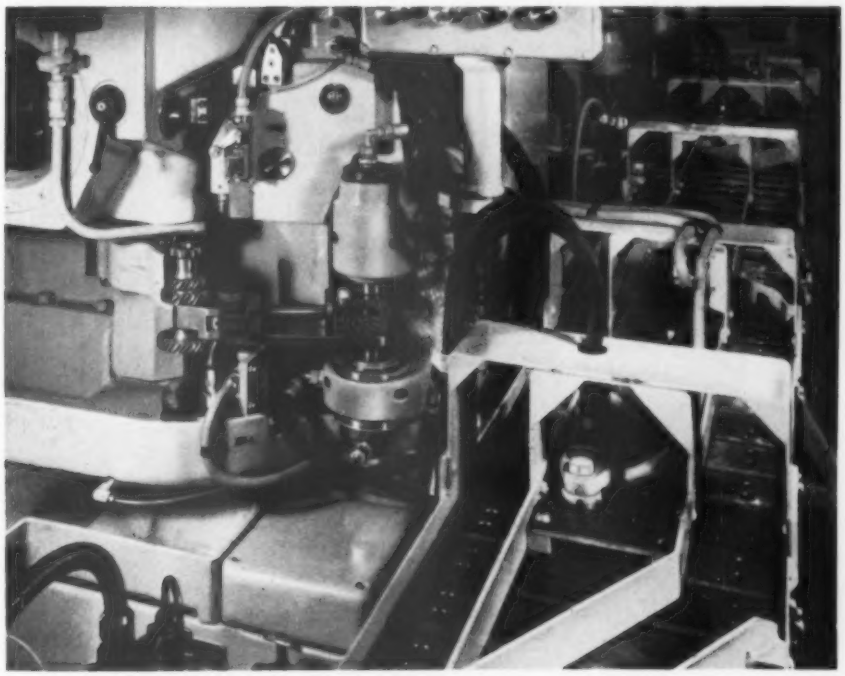
When this gear has been cut, the component is returned from the hobbing machine to the right-hand channel and, by-passing the other machines, is carried past the last of the fourth-speed gear hobbers to the central channel feeding the three hobbers for the third-speed gear. A solenoid-operated gate mechanism temporarily halts the passage of these machined components along the right-hand channel when there are blanks crossing their path while being transferred from the conveyor to a hobber. The storing and by-passing process is duplicated for the machining of third-speed and first-speed gears, and the hobbing machines for these have maximum storage capacities of 101 and 48 components respectively.

At the end of the hobber conveyor the clusters, now back in the central channel and still standing on end, are halted by a gate and allowed to pass through it at intervals of 35 seconds. As the slats of the conveyor pass over the end roller, each cluster falls horizontally into a shaped orifice in a circular plate, but is prevented from passing through by a platform beneath the plate. The plate turns through 90 deg about a vertical axis and the cluster falls on to a ramp, where it is held for a short period while the excess cutting oil drains off. The pneumatic operation of both the gate and turn-round unit is controlled by valves that are tripped by cams on the driving spindle of the conveyor roller.

Each cluster now rolls down a chute and is conveyed slowly through a washing cabinet. Here, a high-pressure system sprays warm detergent fluid over each component for a period of 20 sec, to remove any adhering oil and swarf, which could affect accuracy of location between work centres. An 8 sec blast of cold air dries it off, and the com-



A cluster gear blank has been moved by the preload arm from the storage bank to the inner channel of the conveyor. It is about to be picked up by the jaws of the transfer arm and placed between the hydraulically actuated work-centres of the Churchill hobbing machine



Right: In the centre channel of the conveyor is the storage bank for this gear-shaping machine. When it is filled, oncoming cluster gears are temporarily deflected into the channel on the right, and pass to the next machine. The component shown being unloaded from the machine will be deposited in the left-hand channel by the transfer arm and then conveyed past the remaining shaping machines in the line

ponent is then tilted on to its end again before being passed on to the gear shaper conveyor, running at a speed of 6 ft/min. The hobber conveyor is 54 ft long, between the centres of the end rollers, and the corresponding length of the shaper conveyor is 35 ft. Each is driven by a ½ h.p. motor, through a clutch and gearbox with a reduction ratio of 750 : 1. The function of the clutch is, of course, to slip, in the event of jamming of the component.

Loading and operation of hobbers and shapers

A feature that has been used extensively in conjunction with Churchill hobbers in the past four years is the C.G.M. hydraulically operated transfer arm, and this is now applied also to the gear shapers, since both types of machine have vertical work axes. When the leading component in a storage bank is conveyed forward into the preload arm, a micro-switch is tripped which moves the arm from the central channel to the pick-up station in the right-hand channel. A time switch is then brought into operation to close the jaws of the transfer arm round the cluster gear, then raise the arm and swing it over to the loading position on the machine. Meanwhile, the preload arm returns to the central channel, ready for the next blank.

Movement of the tailstock ram trips the safety switch that controls the start of the machining cycle, so that the jaws of the transfer arm will not relax their grip and the machining operation will not begin if the component is not aligned between the centres. At the end of the cutting cycle, the work spindle stops, the jaws grip the component and the tailstock retracts, in doing which it operates the safety switch controlling the return of the arm to the unloading position above the right-hand channel. Another switch is operated by this movement and the arm is lowered, and in doing so actuates a third switch, which opens the jaws, to release the component into the right-hand channel of the conveyor.

In this arrangement of a production line, several machines perform identical operations, and each is equipped with a fully interlocked system of limit- and safety-switches. All machines are served by a continuously moving conveyor that embodies devices for by-passing loaded machines and also ensuring the prevention of overloading. Thus there is a minimum of interruption to work flow when servicing or tool-changing is required on a machine, and inspection can be carried out at any point in the line. Each of the machines is switched off until a component moves into the fork at the end of the preload arm, to start a loading and machining cycle.

The hobbing machines are standard Churchill products. Of these, the first three, which hob fourth-speed gear, each have a cycle time, including 15 sec loading, of 1 min 45 sec; the cycle time of the three third-speed gear machines is 1 min 30 sec, and that of the remaining two, which hob first-speed gear, 1 min 15 sec. The hob speed is 310 r.p.m. and the feed is 0·060 in/rev. Each machine incorporates an automatic axial traverse of the hob, the life of which is thus increased : the maximum traverse is 2½ in. It is estimated that a typical hob life will be 5,000 cuts : this is based on 20 regrinds and 250 pieces per grind. To warn the operator that a hob is due for either regrinding or replacement, a micro-switch is incorporated to operate a red light when the hob has reached the limit of its axial shift. Changing a hob takes between 15 and 20 minutes.

There is oil mist lubrication, fed by air at a pressure of 10 lb/in^2, for the main drive bevel gears and change gears of each hobbing machine, and all slides have one-shot lubrication fed by air at 50 lb/in^2. In the event of failure of the air supply, a pressure switch is brought into operation to stop the machine. An emergency stop button is positioned for easy operation, and if the button is depressed during a cycle, the hob slide returns to the start position and the main drive motor stops. A modern feature of the rapid traverse motor is that it is fitted with a disc brake instead of a conventional shoe brake.

After being washed and dried, the clusters pass to the conveyor that serves the high-speed shaping machines. The five storage banks on this conveyor can hold 16, 28, 56, 84 and 112 blanks respectively, a total of 296. All five of the machines are engaged in cutting second-speed gear, and the cycle time for each is 4 min 21 sec, including 15 sec loading time. The cutting rate is 417 stroke/min, and each cutter has a life, between regrinds, such that it can shape 100 gears. Apart from minor differences in the methods of mounting the transfer arms on these machines, the loading procedure and interlocking safety devices are identical with those on the hobbing machines. However, special arrangements have had to be made to adapt the shaping machines to hydraulic control of both the transfer arms and the tailstocks.

At the end of the shaper conveyor there is another gear washing and drying unit, operated in the same way as the previous one. Unlike the arrangement with the earlier line at the B.M.C. factory, there is no automatic gauging system incorporated at this point. Instead, manual gauging is carried out by inspectors.

Individual machine loading

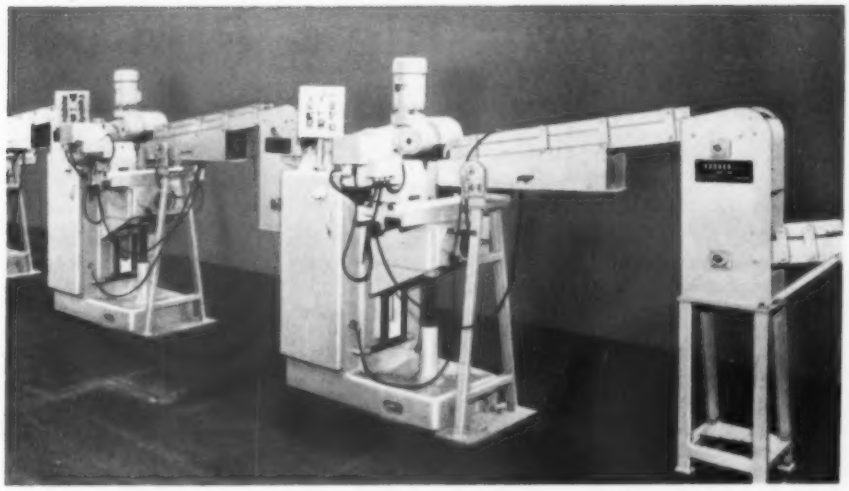
From this point onwards, each of the machines, with the exception of the grinder, is fed by an elevator-ramp-loader system of Churchill design and manufacture. Components, with their axes horizontal, are fed into an elevator which carries them to the first chamfering machine. The elevator is driven by a motor of ¼ h.p, through a clutch, which again is to avoid damage to the mechanism in the event of jamming. At the lower end of the chute, each cluster is picked up by a C.G.M. automatic loader consisting of two pairs of hydraulically operated fingers, and placed between the work centres of the first chamfering machine.

The feed chutes act as storage banks, for they incorporate micro-switches that are operated when the chutes are either empty or contain ten clusters, so that subsequent or previous machines are switched off, as appropriate. These storage banks give the degree of latitude in operation that allows maintenance work or tooling changes to be done on one machine without the need for stopping others. On completion of the chamfering operation, the centres withdraw, and a lever rises to support the cluster, which rolls down another chute to the bottom of the next elevator.

There are nine automatic loaders in the line, one to each machine that is working on a horizontal axis. Seven of these, serving the chamfering and shaving machines, are hydraulically operated, and the two that serve the tooth-rounding machines are pneumatically operated. The loader for the Newall grinding machine is of Newall design and pneumatically operated, and is similar in operation to the Churchill transfer arms used on the hobbing and shaping machines, though, of course, handling the component horizontally instead of vertically.

Chamfering and tooth-rounding operations

The gear chamfering and tooth-rounding machines were designed especially for operation in conjunction with the automatic loaders. There are three chamfering machines, one for each of the helical gears on a cluster, and two tooth-rounding machines, one for each end of the straight spur gear. In the chamfering machine, the cluster is gripped between pneumatically operated centres and the gear is meshed in backlash with a rotary cutter which moves rapidly from side to side with a pneumatically-powered traverse of 0·030 in; no cutting oil is used in this operation. Power for



Two of the tooth-chamfering machines and their loading mechanisms. Components are carried on small elevators to the tops of ramps which act as storage banks. Micro-switches are fitted to these ramps to switch off preceding or subsequent machines when the banks are either full or empty, as appropriate

the machine is supplied by a motor of 1 h.p, and the speed of the cutter is 100 r.p.m. The cycle time for each of the chamfering operations is 22 sec, and the loading time 15 sec.

As in the chamfering machines, the work-centres of the tooth-rounding machines are pneumatically operated. A small spur gear meshes with the first-speed gear on the cluster and rotates it, while the work-table is oscillated in line with the gear teeth and at a frequency corresponding to the passage of teeth past the cutting tool. The path of the tool, and therefore the shape of the ends of the teeth, is dictated by the form of the cam which controls the oscillations of the table.

This cutter rotates at 1,050 r.p.m. and the machine is driven by a 1 h.p. motor; a soluble cutting oil is used in this case. The cycle time is 30 sec, and the loading time is 12 sec. After the second tooth-rounding operation, the clusters move on to a washing and drying cabinet, and are then thoroughly deburred by hand.

Gear shaving and peripheral grinding

In order to adapt the standard gear-shaving machines to automatic in-line loading, their tables are turned through 90 deg, about a vertical axis, relative to the conventional position. The gears are shaved in the order fourth-speed, third-speed, second-speed and first-speed, the number of teeth per gear being 28, 24, 19 and 13 respectively. Their cutter speeds are 193, 193, 137 and 93 r.p.m, and their cycle times are 42 sec, 38 sec, 32 sec and 28 sec respectively.

All the differentially-serrated cutters are 9 in diameter, and are set at an angle of 80 deg to the axis of the work-piece. A table feed of 2·83 in/min is used throughout, and there are four passes for each gear. These passes consecutively have upward feeds of 0·015 in, 0·003 in, 0·002 in and zero. Between regrinding operations on each cutter, 20,000 gears can be cut, and the life of a cutter is reckoned to be 6 regrinds.

When the machining operations are finished, the work centres of the shaving machines are withdrawn hydraulically, and the clusters that leave the final shaving machines pass through the last of the four washing stations before rolling down the chute that feeds the grinding machine. This machine has a stepped grinding wheel and performs a plunge-grinding operation on the peripheries of second-speed, third-speed and fourth-speed gears at one setting. It is equipped with a wheel dressing device which works in conjunction with an automatic compensating system, so that the diameters of the gears are maintained within close tolerances. The frequency of dressing is at the discretion of the operator.

In the case of this machine, the cycle time is 30 sec, and the loading time is 15 sec. After being ground, the clusters roll towards a standard section of conveyor at the end of the gear line, ready for heat treatment and grinding of the bore. At this stage, the checking tolerances on the component are: eccentricity to 0·002 in; profile ±0·0002 in; lead ±0·0002 in per inch face width; and diameter ±0·0005 in.

Automatic Tyre Soaping

PROGRESS in the mass production of road vehicles has seen the gradual elimination of the human element from all but the most basic operations. Automatic machinery has been used for fitting tyres to rims for some considerable time, but there inevitably is some handling to be done. One of the preliminary operations that have invariably been done by hand is the soaping of the tyre beading. Now, however, a machine for this has been produced by The Hymatic Engineering Co. Ltd, of Redditch, Worcs.

It is an automatic soaping unit capable of delivering soaped tyres at the rate of 600/hr, and can accommodate a variety of standard tyres of diameters between 10 and 15 in. In addition, it can be integrated with a tyre and wheel assembly machine, to form a fully automatic line. The tyres roll down a chute, and a release mechanism drops them one by one into a vertical position on to a horizontal pair of rotating rollers: two brushes apply the soap solution to the beading of the tyre as it rotates. Each soaping action takes two seconds, and the tyre is then released for fitting.

German Industrial Standards

FROM time to time we have published notices of translations of German Industrial Standards likely to be of interest to automobile engineers. The German Standards Committee Deutscher Normenausschuss, has for some time had a special department preparing these translations into English. So far, 1,200 translations have been made, and the latest lists can be obtained free of charge from Beuth-Vertrieb G.m.b.H, whose address is Berlin W15, Uhlandstrasse 175.

Spraying Rigid Plastics Foam

Improved Chemical Formulations for Polyurethane Foams Lead to Simplified Application, Reduction of Storage Space and Wider Variety of Uses

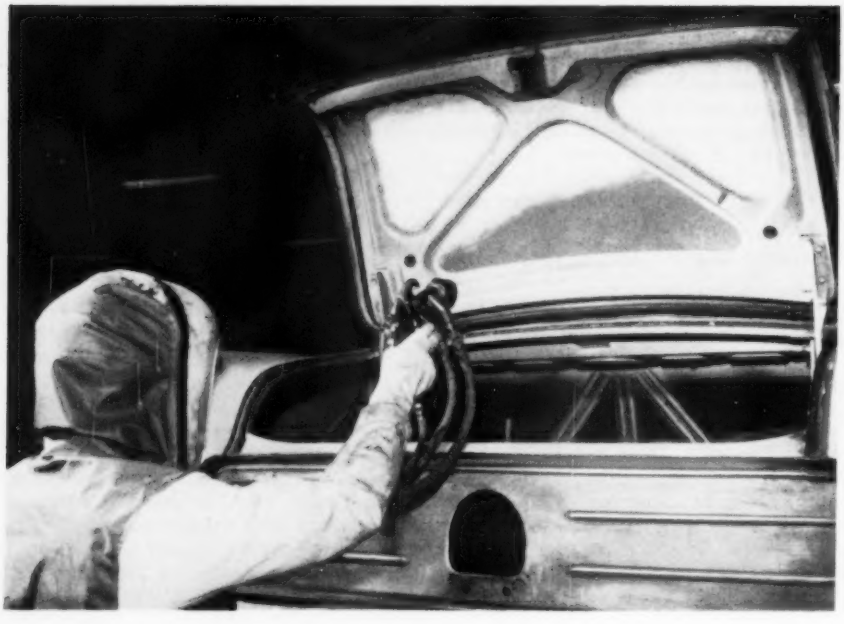
ALTHOUGH the use of rigid polyurethane foam has become commonplace in the spheres of heat insulation and packaging, its use is not as widespread as it might be, because of the expense involved in preparing the mixture, suitably forming the shaped blocks or layers, and fixing them in place. Hitherto, the two methods of producing foam have been machine mixing, from which blocks or sheets are produced for subsequent cutting up, and batch mixing, in which small amounts are mixed by hand, ready for local application. With each of these methods, some form of battening or shuttering is needed to keep the finished foam in position.

Some applications are, of course, best served by such methods of production, but entirely different methods are called for in flow production systems and in applications that would normally require intricately shaped pieces to be fixed in place, possibly in awkward locations. Spraying foam *in situ* is one solution to these problems and it is a method that presents a wide variety of possible applications; until recently, however, it has been impracticable because the two liquid constituents that are mixed to form polyurethane foam were of different viscosities, and it was found difficult to use them in a spraying process unless heat was applied to either or both of them beforehand. Among the firms working towards a solution of this problem was the Baxenden Chemical Co. Ltd, Euston Road, London, S.W.1, in collaboration with The Aerograph-DeVilbiss Co. Ltd, 47 Holborn Viaduct, London, E.C.1.

New formulations have been developed which succeed in reducing the viscosities to a similar low figure. The two constituents are brought together upon ejection from the nozzle of the spray gun and atomized by jets of compressed air : this intermixing produces a Freon-blown polyurethane foam, designated type R 329 "Celspray", which grows rapidly to a thickness of approximately 1 in immediately after being deposited. Greater thicknesses can be obtained by further passes of the spray gun. This foam dries within a few minutes and is completely cured in 24 hours.

Since the foam is not created until it is actually in position, one of the biggest advantages is found in the reduction of storage space required for stock materials. In addition to this, the foam adheres to the surface on which it is sprayed, and there is no need for the application of adhesives or other methods of fixing. If it is desired to make mouldings or to form sandwich insulations, then the

The Aerograph-DeVilbiss type JGC-505 foam spray gun is made of stainless steel. Mixing of the two constituents that create the foam occurs immediately after they leave the central nozzle and the annular one surrounding it. Atomizing air is blown through the surrounding ring of smaller nozzles



Several advantages of foam spraying are demonstrated here. Where applications of plastics foam to car body panels would previously have called for the preparation, cutting and fixing of sheets, these three operations can now be done, in effect, by the spray gun operator. The foam affords a degree of protection against corrosion, and can be painted over

use of the spray process does away with the need for mixing machinery and the attendant troubles connected with the hardening of mixture during down time.

In the factory, service pipes and storage tanks can be lagged very quickly, particularly in awkward corners that would defy conventional methods of lagging, and the treatment of ducting will afford protection against thermal losses and noise transmission from fans. The interiors of packing cases can be sprayed, either before or after the contents have been put in, so that some degree of location is obtained, if desired. Among the uses suggested by the manufacturers is the treatment of vehicle body panels, for noise reduction and insulation against heat and moisture. The only requirements of a surface to be treated are that it is free from grease and dirt contamination and has a working temperature of less than 110 deg C. There is no special technique or skill required, as there is with paint spraying operations, but protective clothing and masks, with their own air supply, must be worn by operators because the chemical action which takes place when the constituents are mixed causes toxic fumes to be given off.

In one of the illustrations is shown the Aerograph-DeVilbiss type JGC-505 spray gun, specially developed for this process. It is made of stainless steel and its nozzle has two main orifices, the central one for projecting the first of the two constituents and a surrounding annular one for projecting the other. Around these again is a ring of small holes, through which the atomizing air, at 40 lb/in^2, is blown. The constituents are fed through hoses from separate 10-gal stainless steel containers, one of which is pressurized directly from a compressed air supply at 100 lb/in^2, and the other through a pressure-reducing valve. In this way, the correct proportions of the two constituents are maintained.

Polyurethane foam has four notable characteristics. The chief one is its value as a heat insulation medium, the K factor being 0·11 to 0·14 BTU/h/ft^2/deg F/in at 75 deg F mean. Its 90 per cent closed cellular construction gives it a compressive yield strength of 15 lb/in^2, but leaves it without as good sound insulation properties as other insulation media. Density is low; the foam has an average weight of 1·6 lb/ft^3, including the outer skin. It affords good protection against erosion and atmospheric corrosion, and can be painted after it has become fully hardened.

Modified Sunbeam Rapier

SUNBEAM Rapier cars now leaving the production line are fitted with an engine of larger swept volume than before—1,592 cm^3 as against 1,494 cm^3. The latest engine is generally similar to that introduced last year for the 1961 Sunbeam Alpine models. Since both the power output and torque are both substantially higher than those of the 1½ litre engine, while the vehicle's weight is only a few pounds more than before, the perfomance should be improved by a worthwhile amount without any sacrifice of flexibility.

The increase of swept volume is obtained by enlarging the bore from 79·0 mm to 81·5 mm, the stroke remaining at 76·2 mm. Apart from this alteration, the larger engine differs in several major respects from its predecessor: as in the case of the Alpine unit, the crankshaft is stiffer than that of the 1½ litre engine, and has larger-diameter big-end journals; the connecting rods and gudgeon pins are also strengthened to withstand the increased loadings. The compression ratio of 9·1 : 1 is the same as that of the Alpine, whereas a ratio of 9·2 : 1 was employed in the 1½ litre engine fitted to both types of car. Owing to the scale effect, of course, it is usual for the optimum compression ratio to fall with an increase of bore.

The choke diameter of the two Zenith 36WIA2 down-draught carburettors has been increased from 28 to 29 mm; a corresponding increase of 2 mm accompanied the enlargement of the capacity of the Alpine engine. Because the Rapier is intended to have less sporting characteristics than the Alpine, there is no alteration in respect of their differences in manifolding and air filtering arrangements. The manifold of the Alpine is fabricated from steel tubing and is of the double siamesed type, in order to provide the minimum of obstruction to the gas flow, whereas the Rapier has a cast iron exhaust manifold, with a single, medial off-take. Each carburettor of the Alpine power unit carries a small air filter of the pancake type, but on the Rapier there is a common oil bath filter mounted above the engine.

These two differences are responsible for the fact that the performance figures of the new Rapier engine, although superior to those of the 1½ litre version, are less good than those of the current Alpine model. The figures for the net power output, torque and b.m.e.p. of the three engines, in the order just mentioned, are as follows: 75 b.h.p. at 5,100 r.p.m, 73 b.h.p. at 5,400 r.p.m. and 80 b.h.p. at 5,100 r.p.m; 88·2 lb-ft at 3,900 r.p.m, 83 lb-ft at 3,500 r.p.m. and 94 lb-ft at 3,900 r.p.m; and 137 lb/in^2 at 3,900 r.p.m, 137·3 lb/in^2 at 3,500 r.p.m. and 146 lb/in^2 at 3,900 r.p.m.

Among the other changes to the car is the adoption of the clutch assembly already employed on the Alpine unit. Though still of Borg and Beck manufacture and having a diameter of 8 in, this assembly differs from that previously installed in being of the strap-drive type and having a hydrostatic slave cylinder with self-adjusting properties. To ensure that the suspension characteristics match the higher engine performance, the roll stiffness of the front suspension has been increased, by the use of an anti-roll bar of ¾ in diameter instead of the previous ⅝ in. As hitherto, saloon and convertible models of the Rapier are available, and Laycock-de Normanville overdrive is an optional extra.

Spire Speed Nut Manual

THE Spire Speed Nut Manual has been revised, and the new issue can now be obtained from Simmonds Aerocessories Ltd, a member of the Firth Cleveland Group. This latest edition, of course, includes all the recent developments. There is a pictorial index at the front and a numerical index has also been included. Tables are given showing the weight per thousand of each part and also some useful conversion factors. Many other useful details, including finishes, recommended installation torques, ultimate tensile loads and screw thread dimensions are incorporated. A feature of the new manual is the inclusion of fully dimensional engineering drawings of each component.

Spaces have been left for new development reports, which will be distributed to all who return addressed cards to be found under the front cover. These reports will be on gummed paper so that they can be stuck in the appropriate space on the page allotted to it. For the convenience of overseas customers, editions of the manual have been prepared with the introductory matter and specifications in French, German, Italian, Finnish, Norwegian, Dutch and Danish. Metric dimensions are given throughout all editions. The address of Simmonds Aerocessories Ltd. is Treforest, Pontypridd, Glamorgan, and of the Export Sales Office is Stornoway House, Cleveland Row, London, S.W.1.

Preforming Blanks for Cold Forging Operations

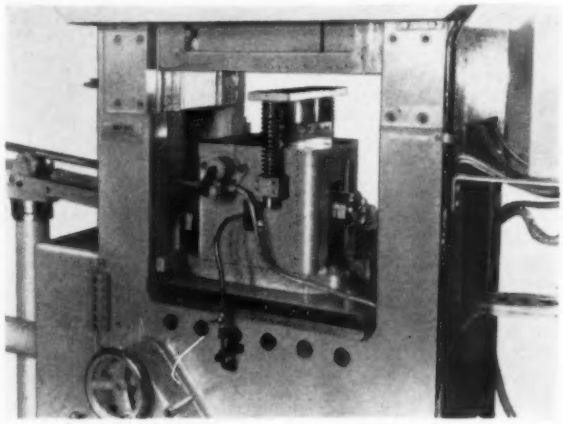
A Conveyorized Machine Line for the Continuous Automatic Production of Uniform Blanks from Rough-rolled Bar Stock

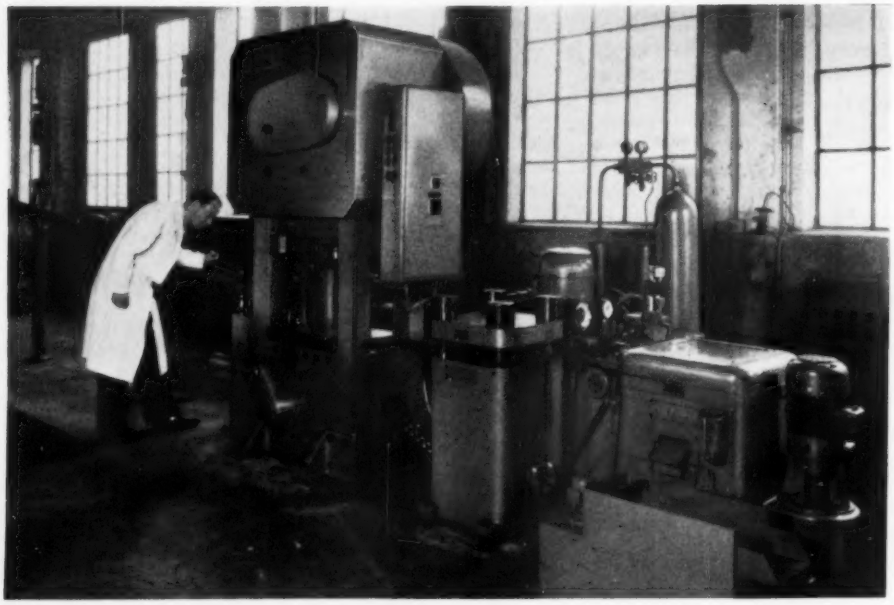
AN essential requirement in the mass production of small components by the cold forging method is an adequate and uninterrupted supply of steel blanks of uniform mass and shape. Three separate operations may be necessary in the preparation of the blank: cropping, flattening or preforming, and deburring or chamfering to remove the burr formed by the cropping operation. Cold Forging Ltd, of Hanworth Road, Sunbury-on-Thames, Middlesex, a company that builds all machine tools and processing equipment required for cold forging, has designed and developed a completely mechanized line for the production of blanks. It consists of a bar feed unit, a cropping machine, a flattening or preforming press, and a specially designed chamfering machine. These units are linked by a single mechanical conveyor that transfers the blanks sequentially from one machine to the next adjacent. This aggregate serves as the first element in the layout of a complete Colforg installation for cold forging small components at high production rates, comprising blank preparation, annealing, descaling, phosphating, lubricating, and press forging. The process of cold forging steel has been described in earlier articles.*

A complete preparation unit, installed at the Company's works at Sunbury-on-Thames, is illustrated. Except for the loading of stock material on the bar stock feeder it is fully automatic in operation. As far as possible, it is constructed of standardized tools and parts so that it can readily be adapted to meet specific requirements.

Cropping machines are built in three sizes, capable of handling bar stock of 1 in, 2 in, and 3 in diameter respectively. In the interests of overall operational efficiency it may be desirable to employ the cropper with a special tool that, additionally, will also perform the flattening operation. The power required to flatten a blank of specific diameter is substantially greater than that needed for cropping bar stock of the same diameter. It follows, therefore, that the machine can be used for combined operations only on stock of smaller cross-sectional area than is rated as its maximum capacity as a cropper.

The bar feeder is a standard unit that serves the machine

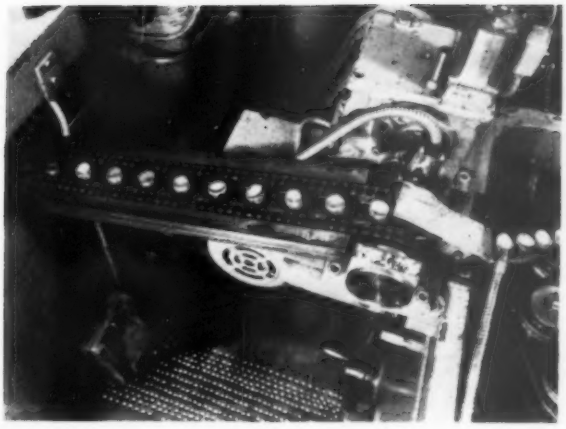




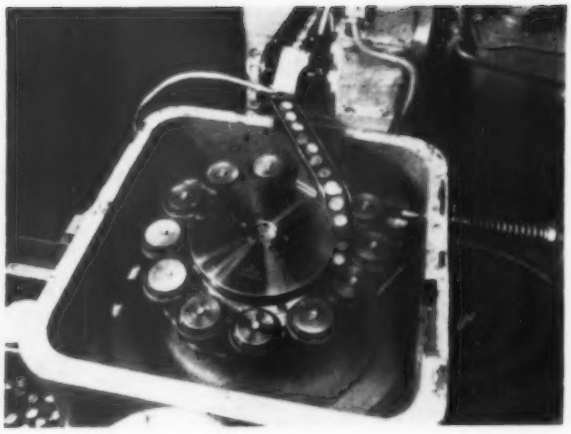
Above: Readily interchangeable tool unit fitted on the bar cropping machine

Automatic cropping, flattening or preforming, and chamfering aggregate for the preparation of blanks, as installed at the Sunbury works of Cold Forging Ltd.

* *"Automobile Engineer", February 1959 and December 1960*



Duplex chain conveyor linking the flattening and chamfering machines



Chamfering machine with cover lifted, showing cutter head and feed

whether it operates exclusively as a cropper or as a combination cropper-flattener. Bars are placed on an inclined bed furnished with rails on which runs a feeder carriage attached by a flexible wire cable to a hydraulically operated drum. The carriage engages the free end of the bar and, when tension is applied to the cable, a positive feed is imparted to the bar which moves progressively forward to a dead stop in time with the stroking of the cropper. Hydraulic operation of the feed maintains a constant pressure on the bar and a constant hold-up to the stop, ensuring accurate feeding and cut-off.

A cropped blank is picked up by a transfer conveyor for transport to the next machine, the preforming press. The conveyor system consists of two parallel, positively driven chains fitted with carrier dogs. These dogs on each chain are arranged to face each other, the space between each pair accommodating a cropped blank. By adjustment of the rails on which the chains run, the relative spacing of the chains can be varied to accept blanks of different size.

Preforming is carried out on an up-stroking machine comprising a frame with an overhead platen mounted on four columns. The blanks enter the machine and are flattened or shaped by the upward movement of a ram to a fixed stop. This ram is hydraulically operated from a self-contained drive unit coupled directly to the press. Assembled on the drive unit are the electric motor, hydraulic pump, valves, reservoir, pre-filling cylinder, and the necessary controls and monitoring instruments.

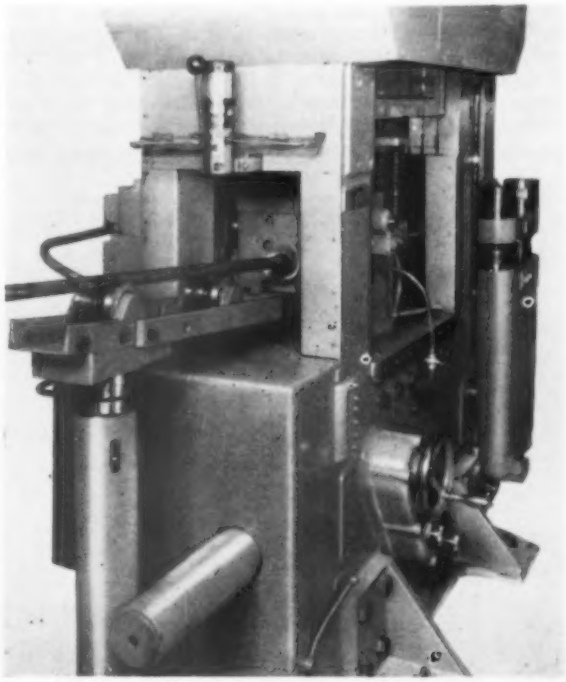
Linking the function of the cropping machine and the preforming press is an electromagnetic clutch control unit driven from the feed conveyor chain on the discharge side of the cropper. This control unit, a standard item that can be used on other machines, actuates the solenoid valves which, in turn, control the operation of the preforming press. The machine can be fitted with a simple flattening tool or equipped with different tools to impart an initial shape to the blank to facilitate the development of the final form in the subsequent cold forging operation.

The conveyor passes through the preforming press and carries the preform to the chamfering machine where it is dropped into a chute. On their return to the cropper, the lower runs of the conveyor chains pass through the pedestal of the preforming machine. At the chamfering machine the blanks feed down the chute by gravity to the working zone and are advanced to the chamfering tool, mounted on a vertical axis, by spring-pressed rollers arranged around the periphery of the tool. Spring pressure, which can be adjusted, is applied by a single, helical spring, mounted externally and tensioning a flexible cable encircling the rollers. Thus, a substantially constant tension is exerted on the rollers but individual rollers are able to move outwardly to accommodate irregularities in the blanks. The stock material used is only rough-rolled bar, it should be noted.

In effect, the chamfering tool is a large, inverted, bevelled milling cutter, and its operating position relative to the specific height of the blanks can be varied by a simple handwheel adjustment. The blanks traverse with a planetary motion through an arc of approximately 180 deg around the periphery of the cutter and then fall to a discharge chute.

As the blank progresses round the tool, the depth of cut varies from zero to a pre-set maximum; the blank supporting track being eccentric to the tool. This method of operation ensures that a uniform amount of chamfer is machined from

Stock entry side of the Colforg fast-stroking, bar cropping machine



the somewhat irregular periphery of the blank. The machine is independently driven by an electric motor. Both the coolant pump and the coolant tank are incorporated in the machine and provision is made for the continuous separation of swarf from the coolant.

Necessarily, there is a break in the automatic flow of material after the chamfering operation. At this stage, the blanks are usually withdrawn from the line for any intermediate heat treatment and for surface treatment—phosphating and lubricating—prior to cold forging. For both of these treatments the blanks are usually handled in batches.

Control of the entire blank-preparation aggregate is from a single, conveniently sited cabinet. Each machine is provided with safety devices that immediately stop all linked units of the system in the event of tool breakage, entry of a defective or malformed blank, or a feeding or loading fault. The arrangement of the aggregate conforms with modern production requirements. Modifications to and extensions of capacity can be readily met by the addition and re-combination of units with suitable transfer mechanism. Individual tools can be operated as completely independent units should they be displaced in any modified arrangement of the line.

Fitting Tubeless Tyres

Hymatic Fully Automatic Machine for Assembling and Inflating Tyres on Wheels in the Vehicle Production Line

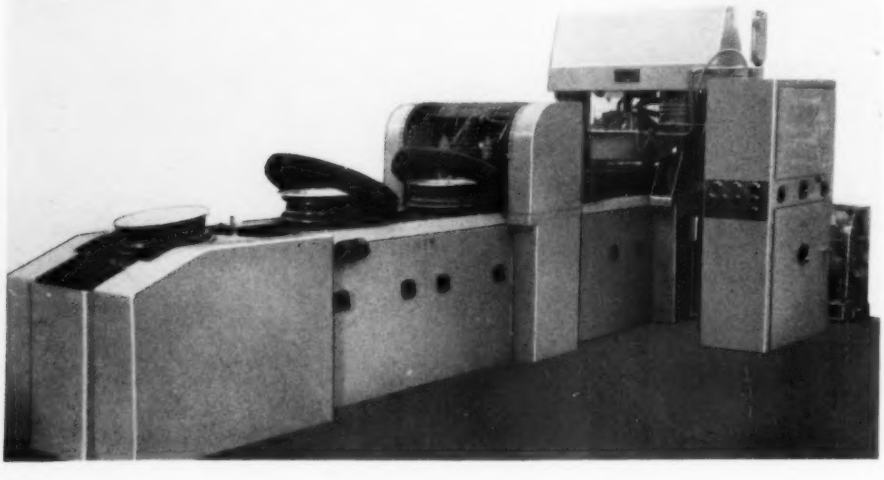
ADOPTION of tubeless tyres as regular production equipment of vehicles called for new techniques in assembly and inflation. Manual methods were too slow and soon semi-automatic machines were made available. Commonly, such machines were detached from the main assembly lines and components were brought to the machine in bulk and assembled wheels were delivered to the production line. An obvious requirement was a fully automatic machine that could be located either in the production line or linked to it by the feeding conveyor system.

Machines of suitable type were developed by the Hymatic Engineering Co. Ltd, of Redditch, Worcester, and have been supplied to several motor manufacturing plants. One such machine, for example, was supplied to the Rootes Group more than a year ago. More recently, a further and improved version of the machine, as illustrated, has been installed at the Rootes Group factory at Ryton-on-Dunsmore, near Coventry.

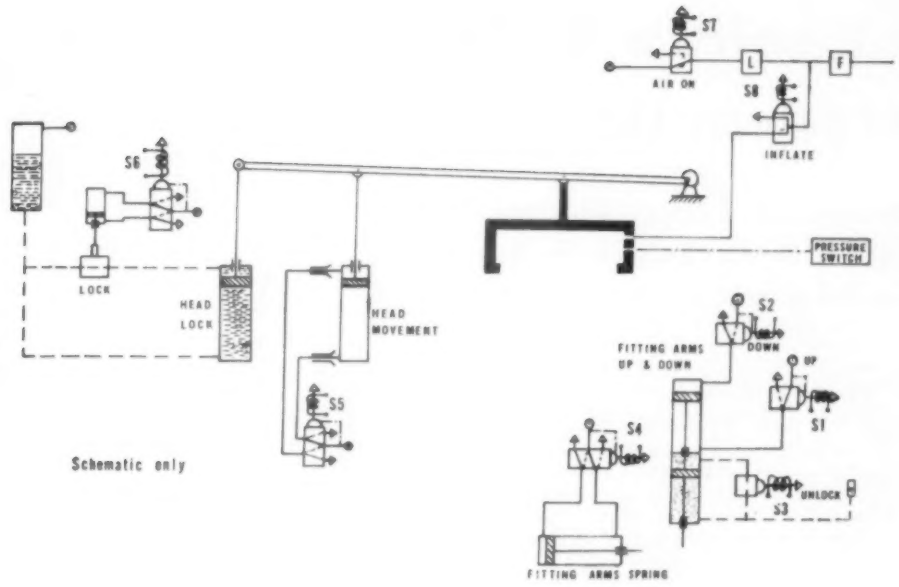
This machine is capable of assembling and inflating tyres, to any specified pressure, at rates up to 450 wheels per hour. An important advantage of the latest model is that it can be used to assemble and inflate tyres of a wider range of sizes than was possible with the earlier machine. Actually it can handle wheels of different rim sizes and the complete range of tyres used on Hillman Minx and Husky, Sunbeam Rapier, Singer Gazelle, Humber Hawk and Super Snipe passenger cars, and Commer Cob and Commer 8 cwt vans.

Wheels are merely laid on the platens of the machine transfer conveyor, which have stepped bases to receive wheels of various different diameters. Tyres are placed loosely at an angle over the wheel rim, with the lower, forward portion of the tyre abutting a pair of vertical pillars on the platen. The conveyor indexes forward to bring each wheel and tyre under a fitting head and, two stations later, under an inflating head. Machine movements are pneumatically and hydraulically operated under electrical control, using cylinders, solenoid valves, and other equipment by Martonair Ltd, of Parkshot, Surrey. No adjustments are necessary to set up the machine for the various tyres of different diameters and widths.

A somewhat simplified pneumo-hydraulic circuit is shown and by reference to this the operational sequence can be followed. When the loosely assembled wheel and tyre reaches the fitting station, two solenoid valves, S_2 and S_3, are energized and a pair of fitting arms press castoring, stepped rollers down on both the wheel rim and the tyre. The two valves are then de-energized and the fitting arms remain locked in position by the hydraulic portion of the fitting-arms cylinder. Solenoid valve S_1 is now energized, but has no effect until the hydraulic portion of the cylinder is unlocked by energizing valve S_4. The conveyor indexes and moves forward and the rollers force the tyre beads over the wheel rim as the assembly passes beneath them. During this movement the fitting arms open and close to follow the



On this Hymatic machine a transfer-type conveyor indexes to carry laid-on wheels and tyres first to a fitting station and then on to an inflating station



Simplified pneumo-hydraulic control circuit for automatic tyre-inflation machine

wheel rim periphery, under the constraint of an "air-spring" cylinder controlled by valve S_1.

When tyre-fitting is completed, valve S_3 operates to allow the fitting arms to rise and the conveyor indexes to bring a previously assembled wheel below the inflating head. This comprises a drum carried on a pivoted beam which is raised and lowered by the head-movement cylinder and can be locked in any position by a hydraulic locking cylinder. As the wheel assembly arrives in position, valve S_5 is energized and the associated air cylinder lowers the inflation head over the wheel. When the head meets the wheel rim, valve S_6 controlling the locking cylinder is energized to secure the head in the seated position.

At this stage the face of the drum casing presses the upper wall of the tyre down to below the level of the upper edge of the wheel rim. Valve S_5 is de-energized, but with no effect until, subsequently, Valve S_4 unlocks the beam. A large inflation valve $S_{7,8}$, controlling a source of filtered and oil-free air, feeds this air at a suitable supply pressure to the inflation drum. The interior of the drum is in communication with the inside of the tyre by way of the annular space remaining between the wheel rim and the tyre wall.

When a specified pressure has been reached in the drum, a pressure switch de-energizes valves S_7 and S_8 and the drum is rapidly raised to expose and clear the wheel with a tyre inflated to correct pressure. The conveyor then indexes and the cycle continues to repeat. Wheels are ejected by the transfer conveyor on to roller tracks for delivery to a shop floor conveyor or to a station from which they are picked off by the forked hangers of an overhead monorail conveyor.

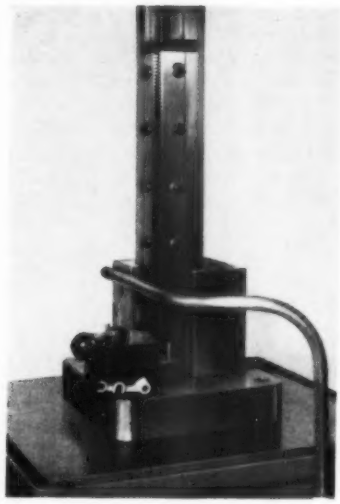
Numerous interlocks are provided on the machine to prevent inadvertent operation in the event of the occurrence of failure to load a wheel or a tyre on a conveyor platen or the faulty positioning of the components on the platen. An interesting example of the type of interlock incorporated to ensure consistently efficient operation of the machine may be quoted. An overriding time switch is fitted to cut off the inflation-air supply if the inflation operation, for any reason whatsoever is not completed in a period of two seconds.

As the loose assembly of wheel and tyre passes through the fitting station, castoring rollers turn the tyre heads over the rim edge

The inflation head is lowered over the assembly, forcing the tyre wall below the rim edge to admit filtered, oil-free pressure air

Lapointe SPD 7/16 Hydraulic Broacher

Bench-height, 3 Ton, 16 in Stroke, Pull-down, Vertical Machine for both Internal and Surface Broaching Operations

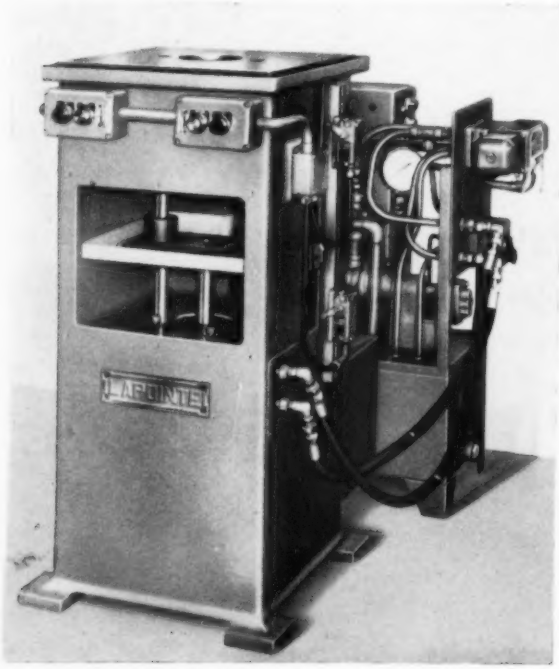


The SPD 7/16 machine arranged for surface broaching two slots in the component shown lying on the workholding fixture

MANY small components that could conveniently and expeditiously be machined by broaching are currently finished by other methods. Until recently, suitable small machines built to modern standards were not generally available and the use of larger, standard machines for such work was economically unattractive. Conscious of this situation, the Lapointe Machine Tool Co. Ltd, Otterspool, Watford By-Pass, Watford, Hertfordshire, conducted a nation-wide survey to ascertain the minimum capacity and the shortest stroke which would provide maximum versatility in this specific field of application in a wide range of manufacturing industries.

An analysis of the Company's sales of broaches showed that 75 per cent of the smaller broaches supplied could be operated on a machine having a 16 in stroke and which could pull a broach of 20 in length. The Model SPD 7/16 was developed from this fact. It was decided to design such a small machine from basic principles rather than to build a reduced-scale version of a standard larger machine. In this way, many of the difficulties likely to be encountered in such a scaling-down process were avoided. The first production machine was demonstrated at the 1960 International Machine Tool Exhibition at Olympia, London, and evoked an immediate response from manufacturers. Capital outlay for the machine, complete with all necessary hydraulic equipment, is considered to be very reasonable by modern standards.

This bench-height, vertical, broaching machine, with 3 ton pulling capacity and a 16 in stroke, is completely equipped with hydraulic power unit and tanks. It requires approximately 12 ft² of floor area



Designated Model SPD 7/16 (the numerals indicate the pull capacity and the stroke) this "bench height" machine is of fabricated steel construction and has a platen only 42¼ in from the shop floor. With the complete hydraulic power unit installed to the rear of the machine the floor space occupied is only 12 ft² — 3 ft wide by 4 ft deep. Incorporated are a 10 gal hydraulic oil tank on the power unit and a 5 gal coolant tank on the machine. The coolant pump is mounted on the power unit. Total weight of the self-contained aggregate is 1,350 lb.

The machine can exert a pull of 7,000 lb (nominally 3 tons) and the maximum length of broach that can be accommodated is 21 in. Maximum stroke is 16 in but the operational stroke can be pre-set to any lower figure to suit the work undertaken. Similarly, the cutting speed of the broach is steplessly variable between 5 ft/min and 15 ft/min. The pull is on dead centre and is constant and the broach pull-head is guided for the full stroke. Platen dimensions are 18 in by 18 in and the platen hole is 5 in diameter. As is usual on hydraulically operated machines, no specific safety device is required. In the event of the stroke being impeded for any reason, a hydraulic valve will blow at a pre-set maximum pressure. Should this valve malfunction the motor driving the pump will stall, and therefore no damage will be done to the machine.

Surface broaching operations can be carried out with equal facility. In view of the diversity of work the necessary equipment is specifically designed for each component. This involves a workholding fixture built as a unit with the broach-body guide, the broach body, and the broach or broaches. An illustration shows a set-up for broaching two slots at 90 deg to each other in a brush holder for an automobile electrical unit.

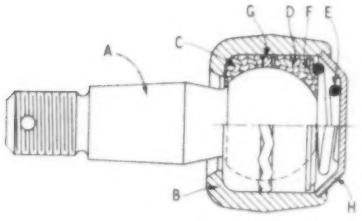
Installation presents no difficulties since the only requirement is to establish connection with an electrical power supply point. Compliance with machine tool, electrical, and hydraulic standards, and the provision of two push buttons to start the broach "cut" stroke, ensure operational safety.

CURRENT PATENTS

SELECTED ABSTRACTS OF RECENTLY ISSUED SPECIFICATIONS

Ball joint

THIS patent is for a steering ball joint that does not need lubrication. The construction of the ball joint is conventional, in that the ball stud A is located in the housing B by two bearing cups C and D, and a degree of pre-loading is imparted by coil spring E that bears against the collar F formed integrally with D. This spring also compensates for wear. An end-cap H is the abutment for the spring, and is semi-permanently fixed, by peening, in the housing B.



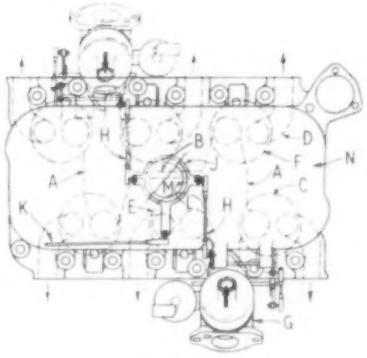
No. 858693

Rubber is the bearing material of the cups C and D, between which is a serrated collar G made of an elastic synthetic material such as a polyamide. The purpose of the collar is twofold: it resists wear and lateral deflection better than would the rubber cups, and by virtue of its serrated form it firmly grips the rubber surfaces. *Patent No.* 858693. *Viktor Langen (Germany).*

Narrow angle vee six engine

A PREVIOUS patent specification has been published* in which a narrow angle vee engine, of an unspecified number of cylinders, had its camshaft situated in the cylinder block, and its cam followers in the head, which was common to both banks of cylinders; the inlet manifold was between the valve rocker assemblies. A new patent taken out by the same company provides specifically for a vee six engine, but the camshaft is now set centrally in the cylinder head, and two carburettors are incorporated, one on each side of the head. Vertical valves are featured, as before.

In the new design, the inlet manifolding is integral with the head, and in the illustration, which is a composite plan view, the chambers and porting are shown by dotted lines. There are two banks of three cylinders, in staggered formation, and each carburettor is arranged to feed two cylinders of one bank and one cylinder in the other. Carburettor G, for instance, is at the entrance of one of the two transverse inlet tracts A; this tract



No. 860011

passes above and between two adjacent combustion chambers C and E, and crosses to feed the chamber F in the opposite bank. Two inlet ports branch off to feed the chambers C and E.

This arrangement is duplicated for the remaining three chambers, but on the other side of the head. There is a longitudinal balancing port B between the two tracts A. An exhaust port D leads from each combustion chamber to separate external manifolds, one on each side of the head. Thus, leading out of each side there are three exhaust ports and one inlet port.

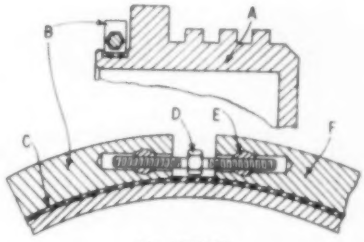
Rocker cover N has a central oil filler in the form of a short vertical funnel M, above which there is a breather cap. A shouldered portion L of this funnel is used as a large diameter pivot pin for lever J in the throttle linkage, which transmits movements of the throttle pedal to the two carburettors as it reciprocates about its vertical axis. There are three arms on the lever, the longest of which is arranged approximately transversely when the throttles are closed, and connected to the longitudinal rod K that leads from the throttle pedal cross shaft; the other two are diametrically opposed on an axis at 90 deg to that of the first lever, are of equal length, and are coupled to the transverse links H that operate the butterfly levers on the carburettors. *Patent No.* 860011. *Austin Motor Co. Ltd.*

Damping brake squeal

IT has been found that squeal from a drum brake can be reduced if a band of resilient material is wrapped round the drum. However, the effectiveness of the remedy can be limited by distortion, heat and wear and tear, and these factors can prevent the elimination of more than part of the total noise. This patent specifies a a resilient band C of silicone rubber, held in place by an extremely rigid steel ring.

The ring is divided into semi-circular segments F and B, and two turnbuckles D are used to clamp them together. These segments are of very wide cross-section, and their end faces are drilled to allow for entry of the turnbuckles. At right angles to each of these drilled holes, and coincident with it, is a larger hole in which is housed the nut E of the turnbuckle. This nut is cylindrical, but its tapped hole is at 90 deg to the cylinder axis; the nut can, therefore, rotate slightly and allow for small variations in alignment of the two segments.

Axial location of band C on the brake drum A is afforded by the sides of the groove in which it rests. This groove is between the rib around the flange of the drum and its open end. Clamping ring F/B, when fully tightened, sinks into the groove to some extent, and is also axially located by it. The extreme rigidity of this ring, coupled with the physical



No. 857043

characteristics of the silicone rubber band, ensure that there is a maximum energy absorption and that there will not be such a tendency for the ring to straighten between the peaks of the wave form caused by drum vibrations as there would be if the ring were of thin section. *Patent No.* 857043. *Aktiebolaget Scania-Vabis (Sweden).*

Cambered wheels on a beam axle

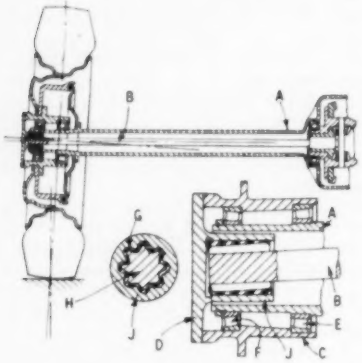
INTERMITTENT lateral forces, such as wind or centrifugal force, acting on a vehicle can upset the handling and directional stability characteristics, so that correcting movements of the steering wheel have to be made if the vehicle is to maintain its chosen course. For these and other reasons, it may be desirable that driving wheels mounted on a beam axle should be cambered or toed-in, or possibly both. Several schemes are suggested which make such arrangements feasible; some of these involve axles that are of a wide V shape, with wheels co-axial with the arms of the V, and others have their wheel axes at an angle relative to the centre line of the axle.

In both types of arrangement, provision has to be made for one end of each half-shaft to transmit drive while its axis is at an angle to that of its mating component,

* *Patent No.* 850129. *"Automobile Engineer", January* 1961.

either a differential gear or a wheel hub. As an example, the illustration shows an axle A whose half-shafts B are not co-axial with the wheel axes. The half-shaft is of the fully-floating type, and a short portion H of its outboard end is coarsely splined.

Wheel hub C is basically of cup shape and rotates on roller bearings E and F



No. 859657

mounted on the axle; the axis of the seatings for the inner races of these bearings is, of course, at an angle to the axle centre line, so as to provide the required camber and toe-in for the wheel. Projecting from the flat end D of the cup is a cylinder J, whose outer diameter is such as to allow ample clearance between it and the bore of the axle casing, and whose inner surface is coarsely splined similarly to the half-shaft. However, between these two splined surfaces there is a layer G, of rubber or plastics, which both transmits torque and allows angular flexure. *Patent No.* 859657. *Ford Motor Co.*

Resilient device acting as a spring

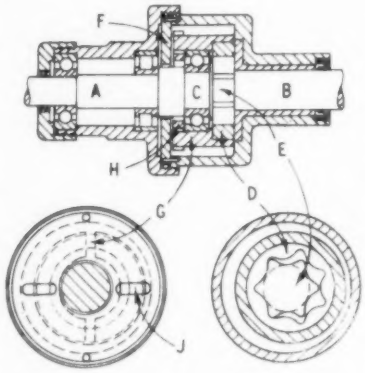
THIS invention is a device intended to replace Belleville washers, coil springs and the like. It consists of a stack of cambered washers mounted on a central spindle and arranged alternately with their rims and centres together. The result is a device that behaves in the manner of a compression coil spring when subjected to axial loads. Each washer is made of a moulded plastics material with elastic properties, for example a polyamide resin. Such a material imparts to the device a high resistance to corrosion, freedom from the need for lubrication in service, and the characteristics of an electrical insulator.

Since the washers are moulded, the spring rate is closely controllable, and it is possible to construct a spring of progressive rate by varying the characteristics of successive washers in the stack. In the illustration is a sectional view, of which the left side depicts the spring compressed, and the right shows it in the free state. The washers B are stacked on a spindle A, and an abutment is formed by a shouldered portion of the spindle. Axial loads are imparted by the sleeve D, which slides freely over spindle A and serves to locate the washers axially when the spring is at rest.

Each washer is in the shape of a dished wheel, its hub having parallel ends and its rim having one plane side and one grooved side. The plane sides of adjacent rims are kept in contact by the presence of a tyre shape ring C that encloses the outer part of each pair of washers, its bead edges registering with the grooved sides of the washers. Rings E are interposed between adjacent hubs, and are of softer material than the washers so that, when the spring is fully compressed, they impart a cushioning effect. *Patent No.* 857611. *Regie Nationale des Usines Renault (France).*

Steering gear

IN the design of this steering gear the main object is compactness. The input and output shafts are coaxial, and the intention is that the steering drop arm, not shown in the illustration, should be fixed directly to the output shaft B. There



No. 856382

is an eccentric spigot C at the end of input shaft A; it is carried in a ball bearing which is housed in a sleeve whose movement is constrained to be gyratory, but not rotary, as will be explained. Secured in the sleeve is an internally toothed gear D, which meshes with gear E on the end of the output shaft B. In the lower right illustration gear D is shown, as an example, to have 7 teeth, and gear E, 6 teeth; so one gyration of D causes $\frac{1}{6}$ of a revolution of E, thereby giving, in this case, a ratio of 6:1.

A form of Oldham coupling is used to impart the gyratory constraint just mentioned. A fixed plate F, dowelled to the gearcase, has two horizontally disposed slots in which slide the lugs J projecting from the movable plate H. Also in plate H are two vertically disposed slots in which slide the lugs G projecting from the sleeve that carries gear D. Thus, rotation of the eccentric C causes gear D to move in a circle but not to rotate about its own axis. *Patent No.* 856382. *George Kent Ltd.*

Differential lock

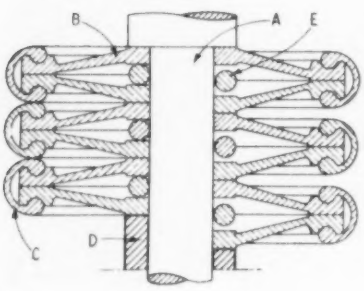
WHEN icy roads or application of excessive engine power might cause one driving wheel of a vehicle to slip, it is desirable to render the differential temporarily ineffective. Described in this patent specification is a hydraulically-operated clutch incorporated in the axle so that one half-shaft can be locked to the differential cage; operation of this clutch is controlled by the driver.

In the illustration, crown wheel B is bolted to a flange of the differential cage C, and is driven by pinion A. Within the cage C are differential pinion D and the bevel gears E and R, which are splined to the inboard ends of half-shafts K and Q respectively. A cylindrical portion of cage C extends outwards to encompass a splined part of shaft K outboard of bevel gear E; the bore of the cage extension is also splined, and the ring shape clutch plates G are contained in the annular space between the extension and the half-shaft. The clutch plates are alternately splined to the extension and the shaft.

The piston H, which imparts the compressive force on these clutch plates is in the form of a ring encompassed on three sides by the cage extension. There is another ring F to form the abutment at the other end of the bank of clutch plates. It is axially located by a circlip in the cage extension, and the axial thrust of bevel gear E is taken by it.

Outboard of the clutch housing, a shoulder on the extension carries one of the taper roller bearings N on which the cage is mounted; the extension continues beyond and inside the roller bearing to finish flush with the edge of the ring P that screws into the casing L to locate the bearing. Drilled passageways J carry oil for actuating the clutch from the external feed pipe M. This oil passes through casing L, ring P and extension C to the annular space in which the piston slides. *Patent No.* 858528. *General Motors Corporation (U.S.A.).*

No. 857611



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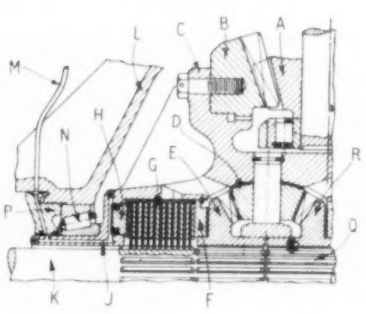

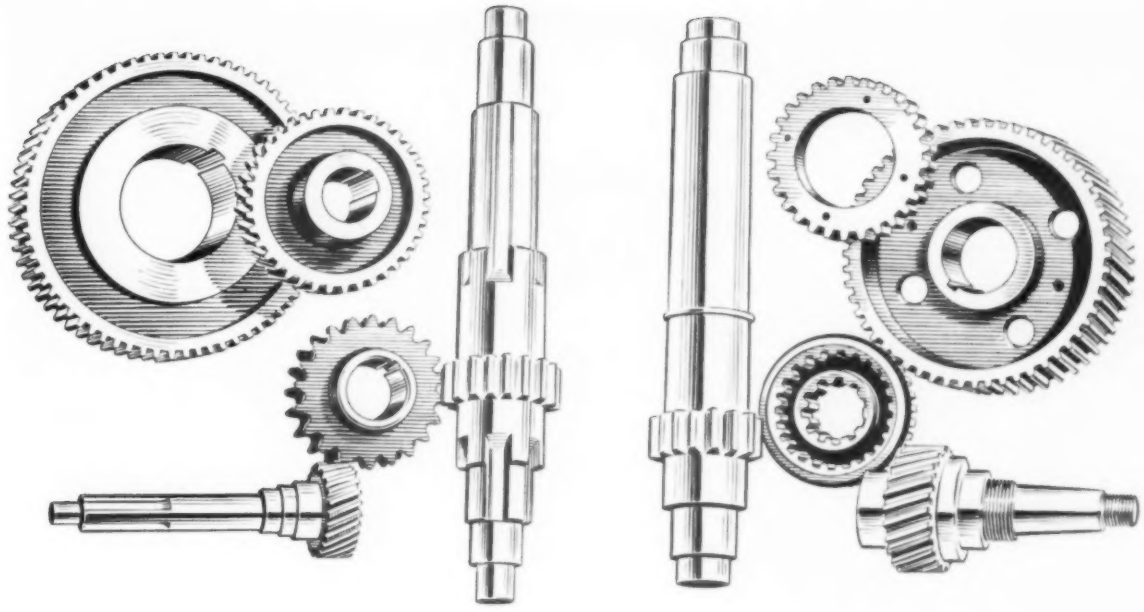

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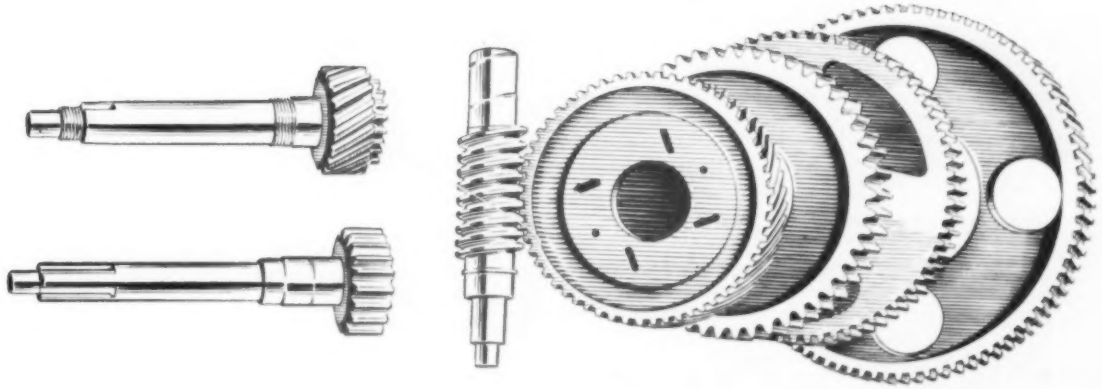

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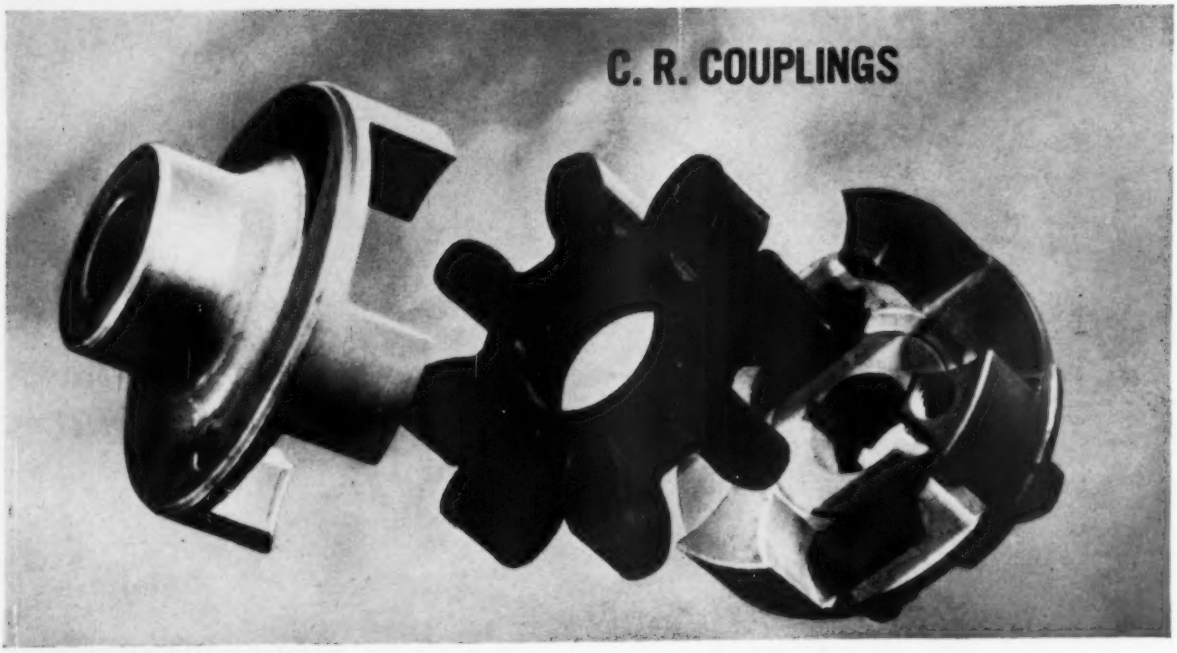

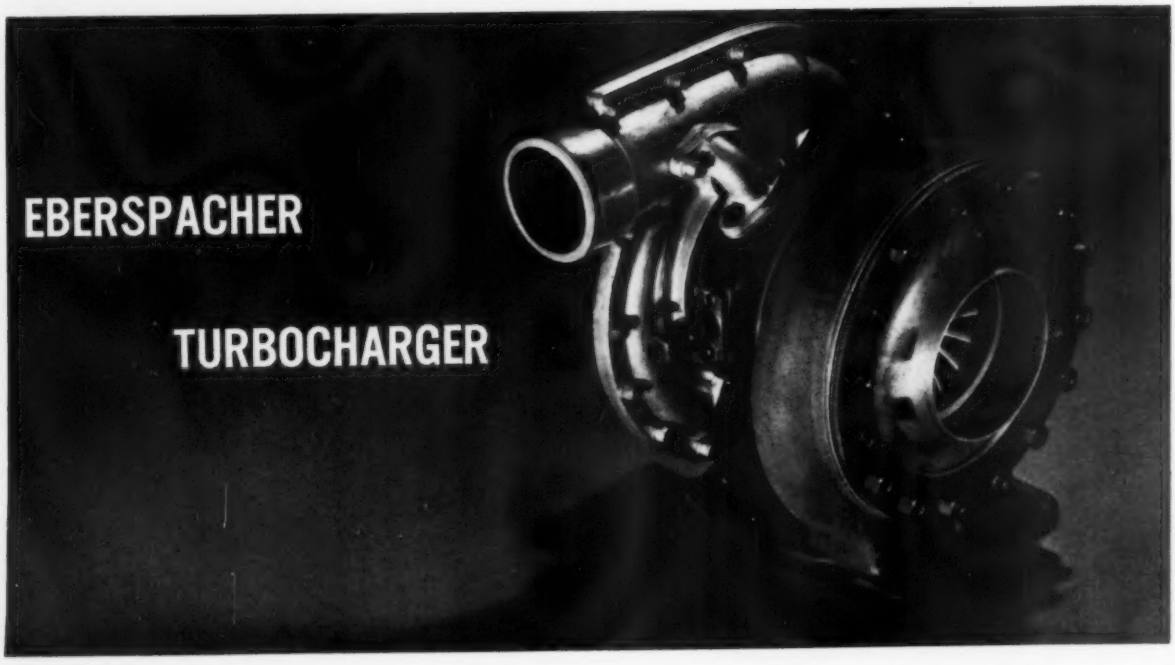

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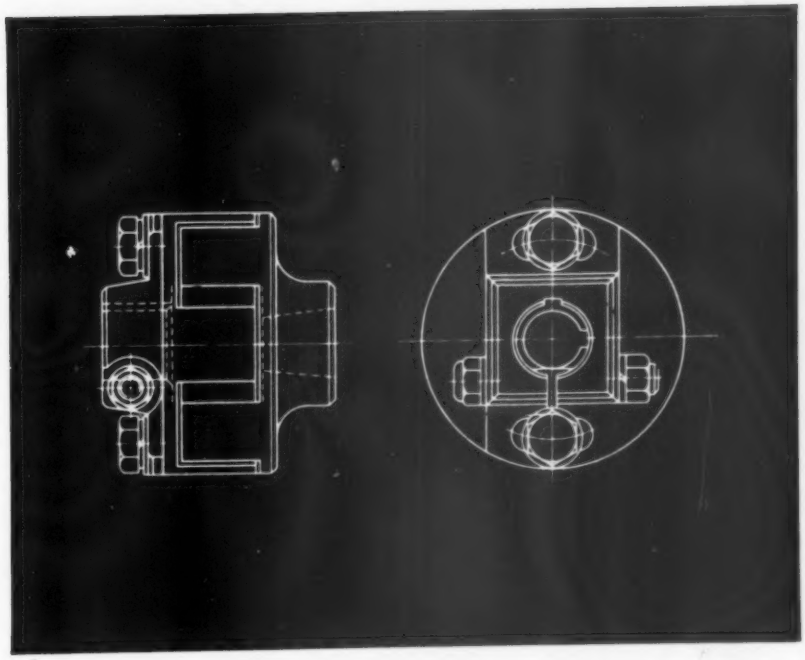

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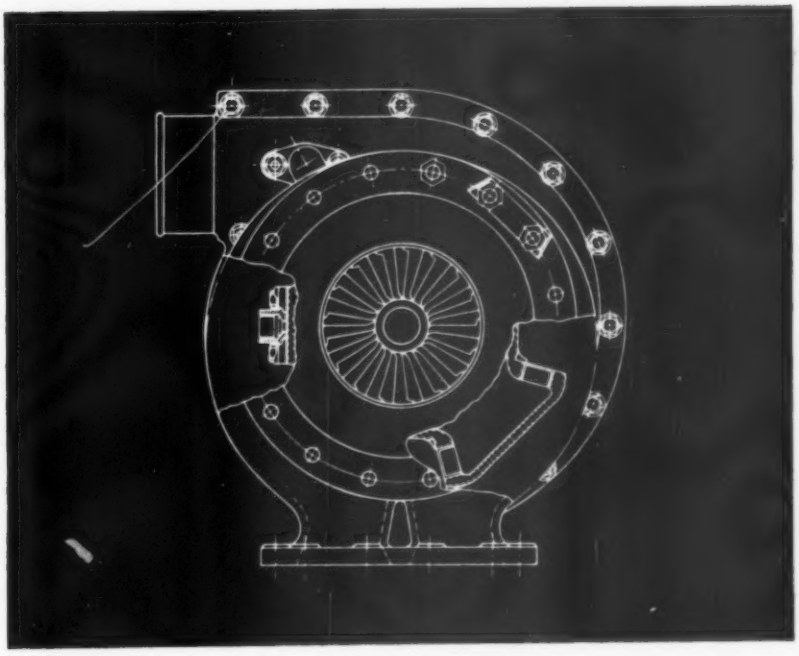

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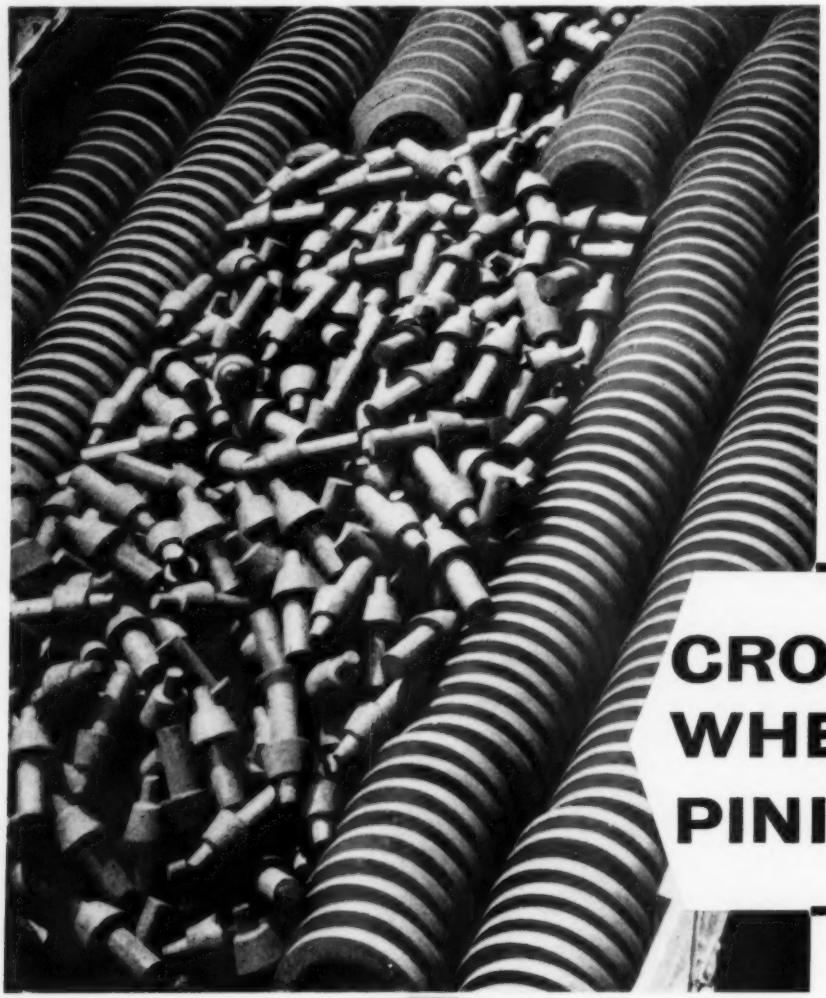

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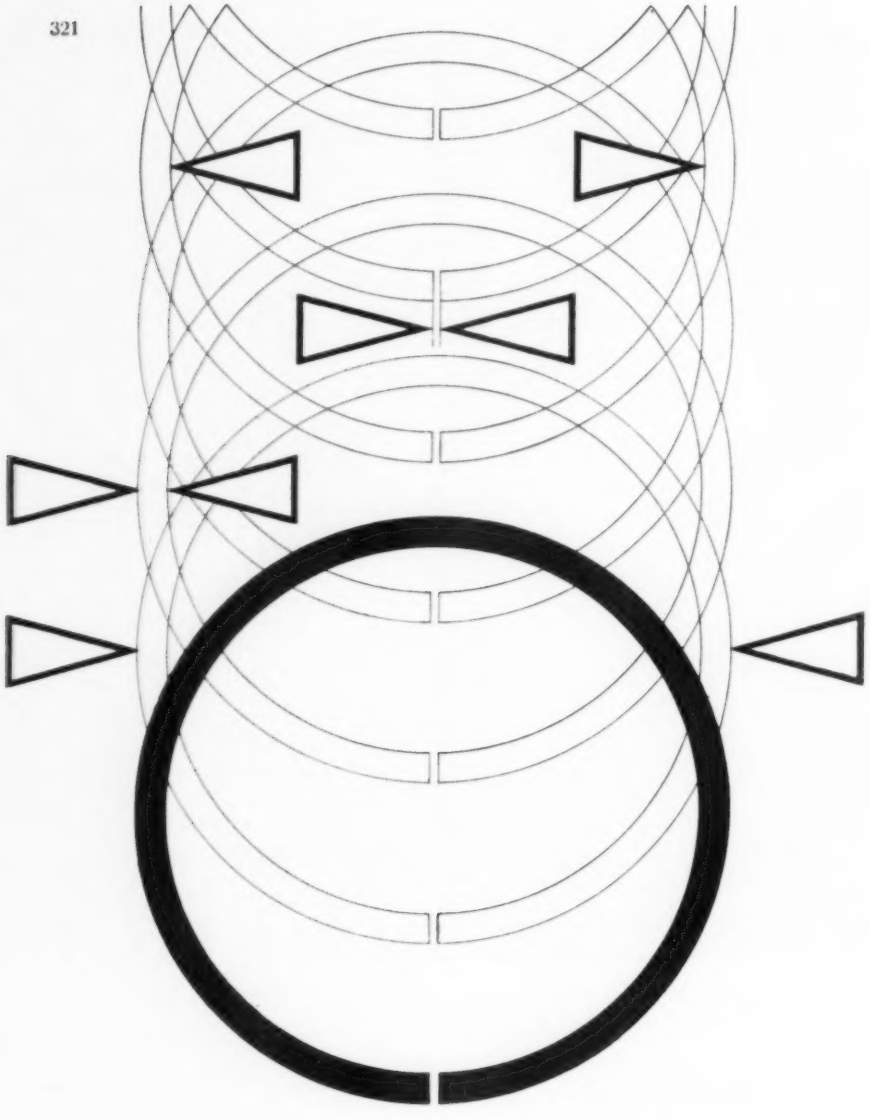

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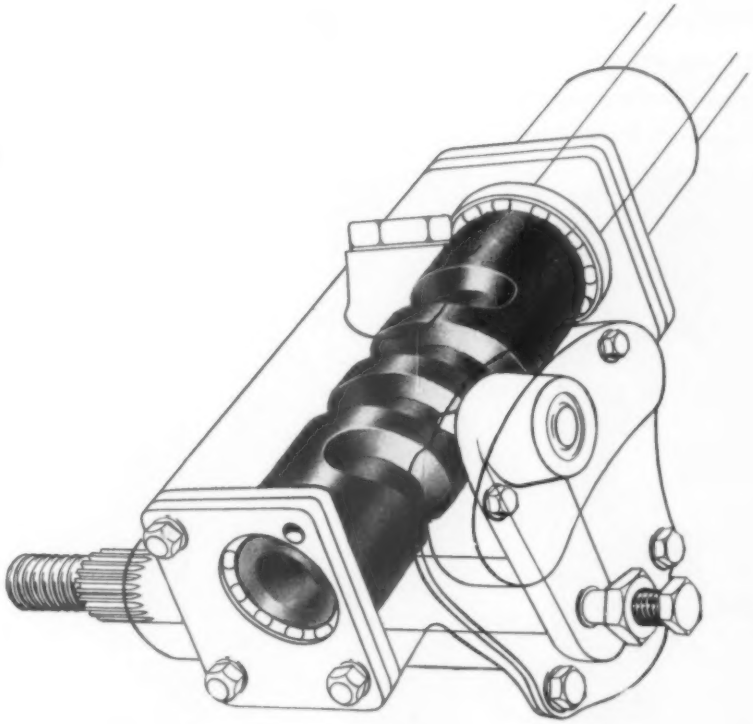

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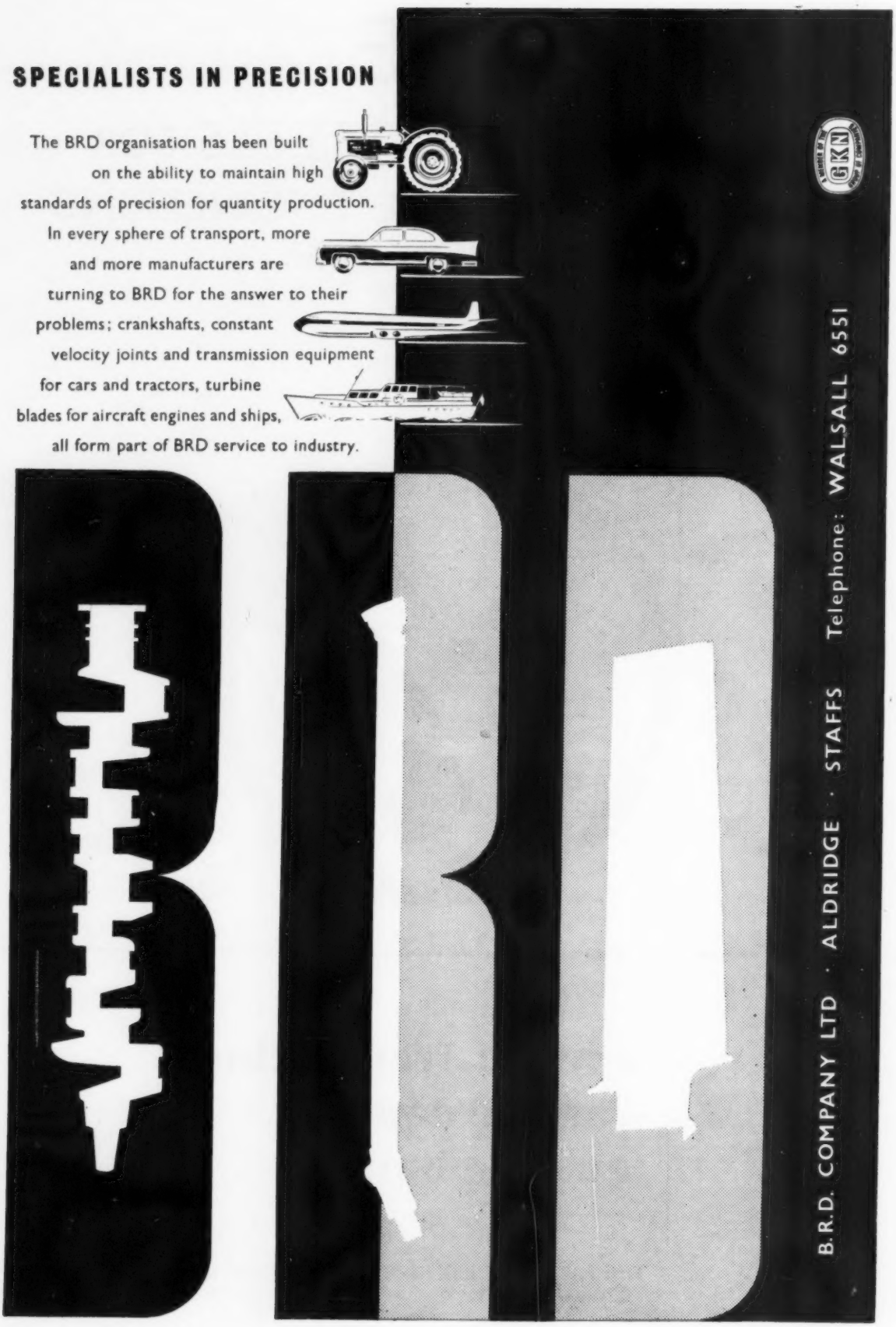

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Principal of the Transport Department

The undersigned, PUBLIC NOTARY AND COMMISSIONER FOR OATHS, CERTIFIES that Mr. JUAN BLUME TRAVERSO, proving his identity with his electoral papers number six hundred fourteen thousand five hundred and ninety four and his military papers number eight hundred and seventy seven thousand six hundred and sixty two, was present in the offices of the aforesaid Public Notary, with the object of having legalised his signature in the DECLARATION which he has signed as Technical Principal of the Transport Department of the Peruvian Omnibus Company Limited, ratifying before me the said declaration which he now repeats and says as follows: "Herewith we wish to certify that Omnibus number two, of "SCANIA VABIS" make, Motor number sixty two thousand one hundred and seventy nine, Chassis number ninety one thousand nine hundred and seventy, which operates on the Lima-Ica Line, has run one million and two hundred thousand kilometres without it being found necessary to dismount the engine for repair", which he signs again in my presence, in Lima, on the thirteenth day of the month of April of the year one thousand nine hundred and fifty seven.

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DR. DANIEL CESPEDES
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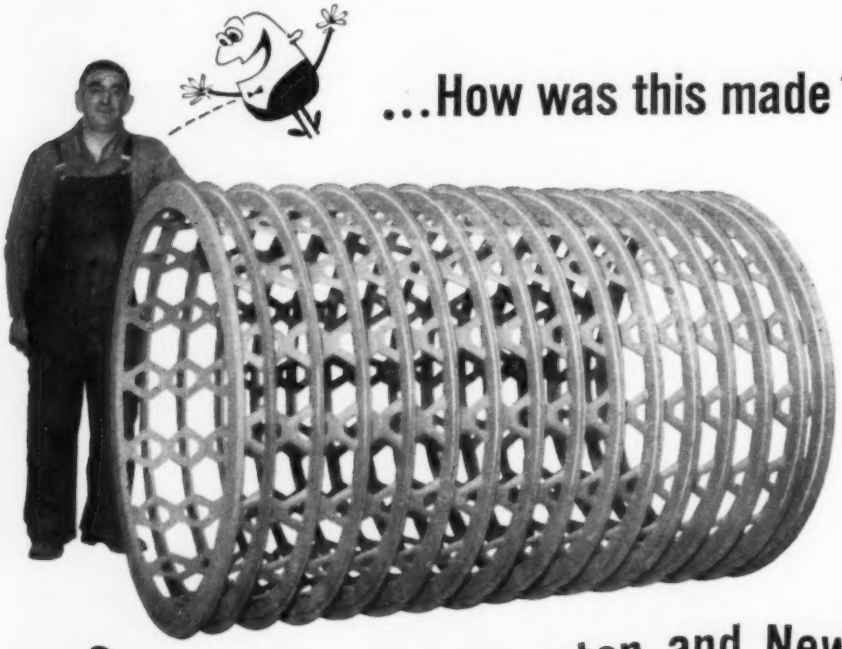

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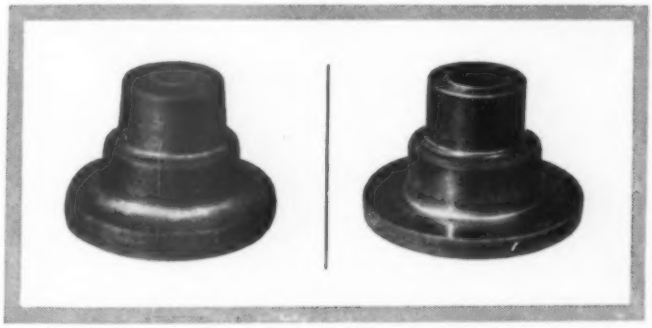

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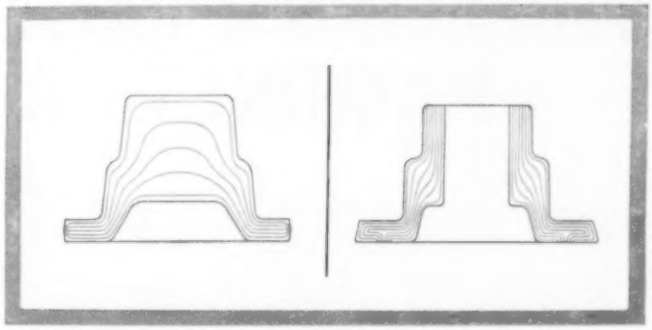

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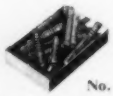

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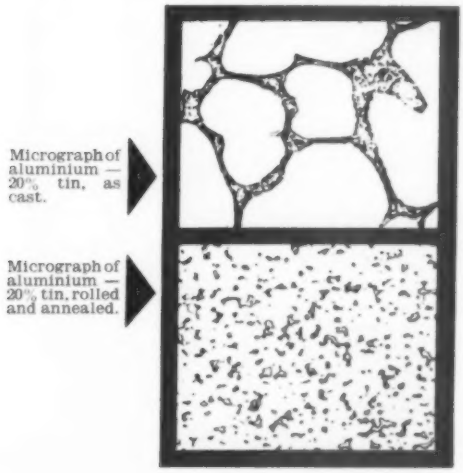

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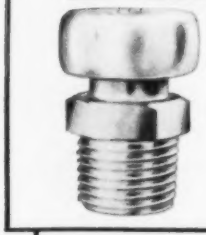

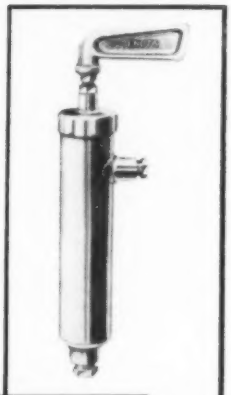

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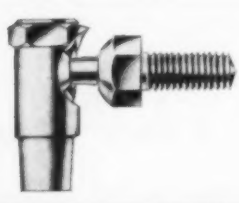

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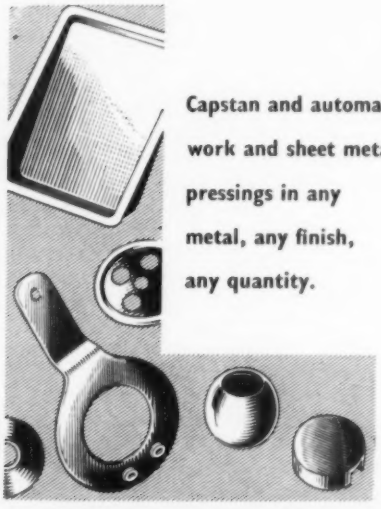

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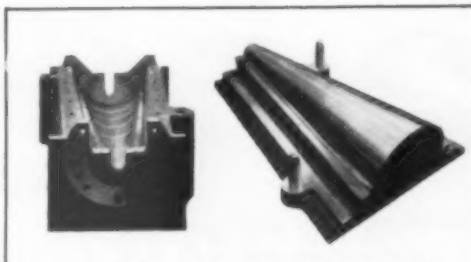

Showing the high finish obtainable on polished tools used for production of six-feet long asbestos components.

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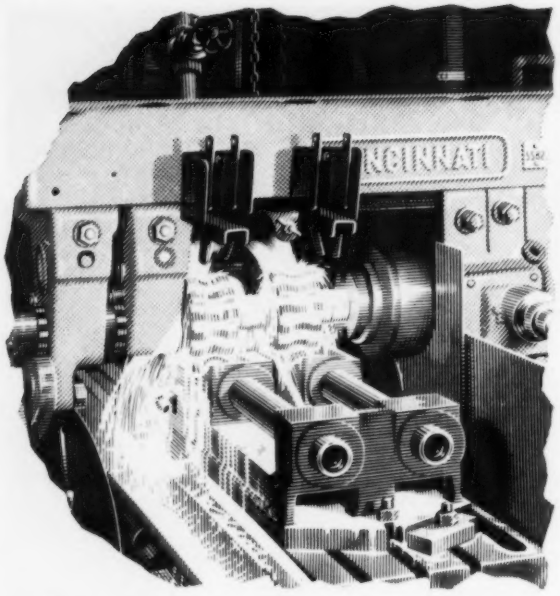

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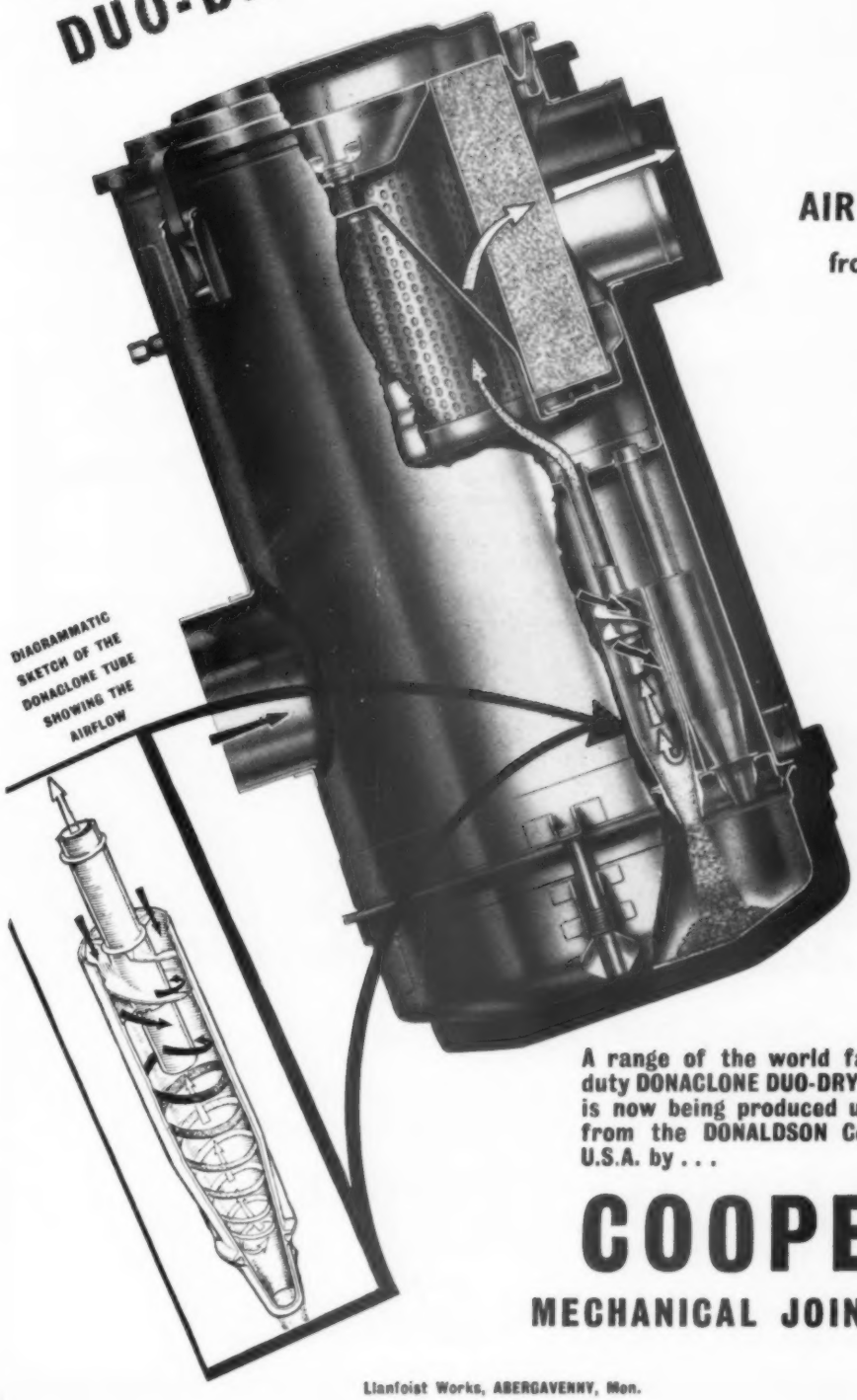

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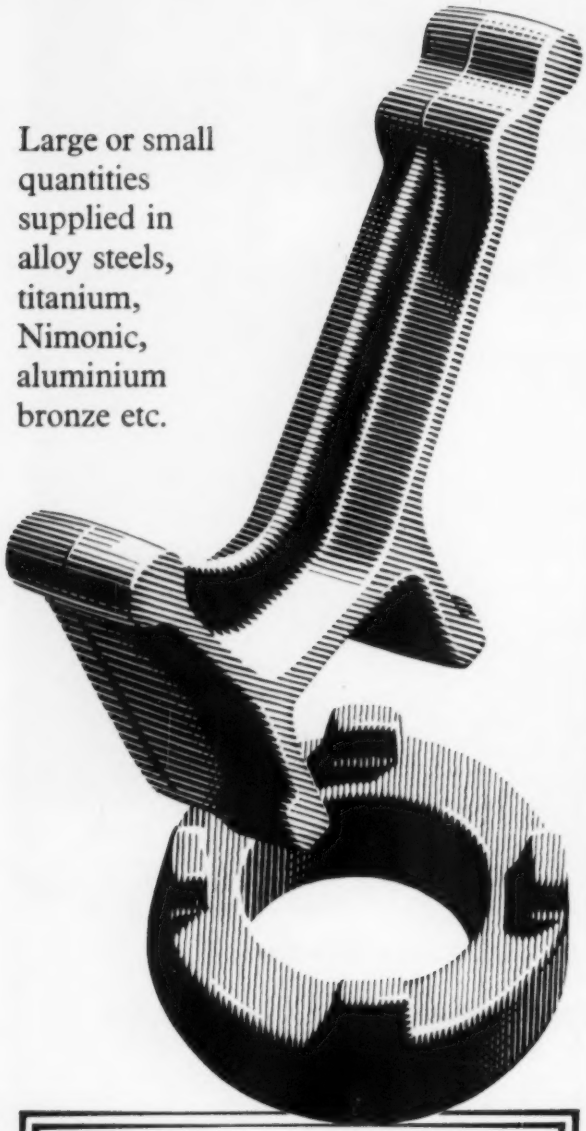

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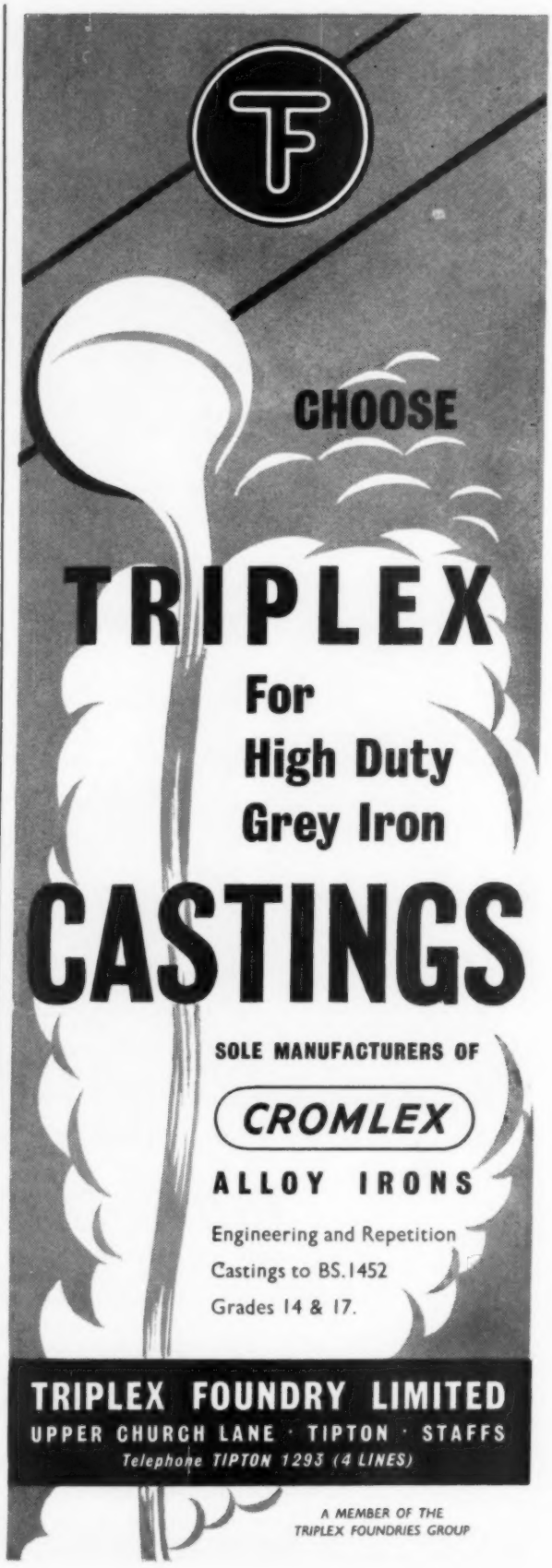

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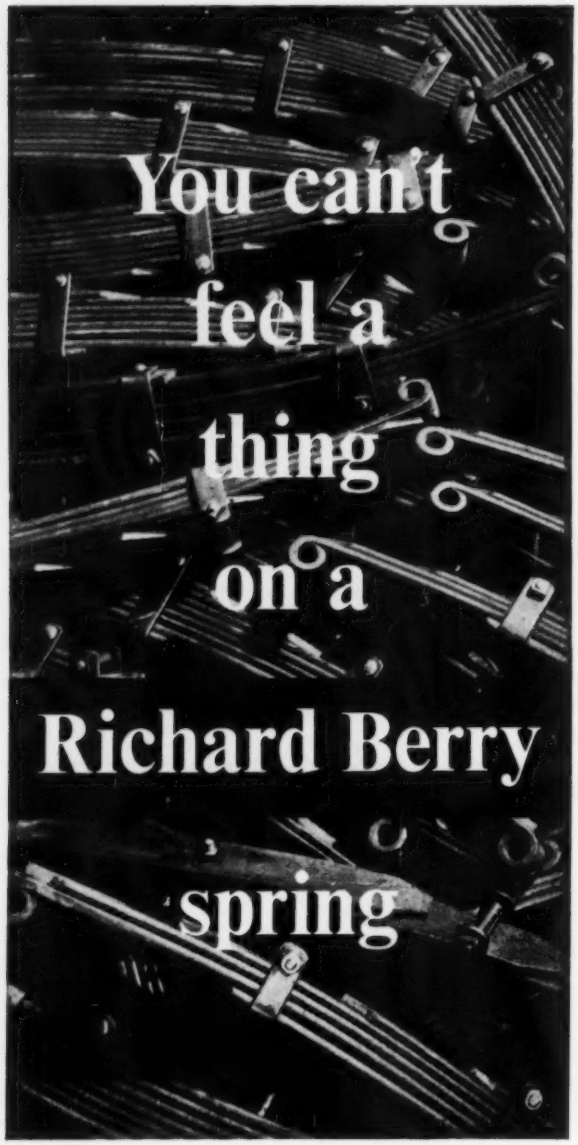

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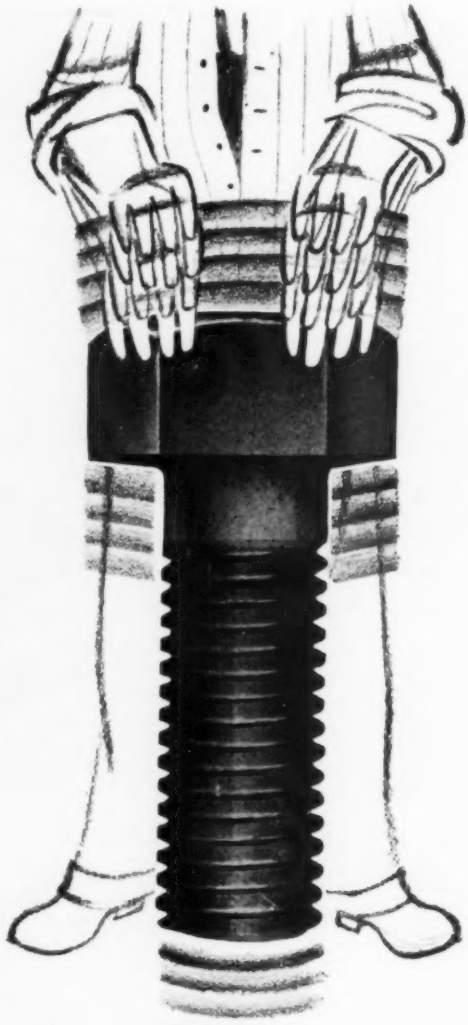

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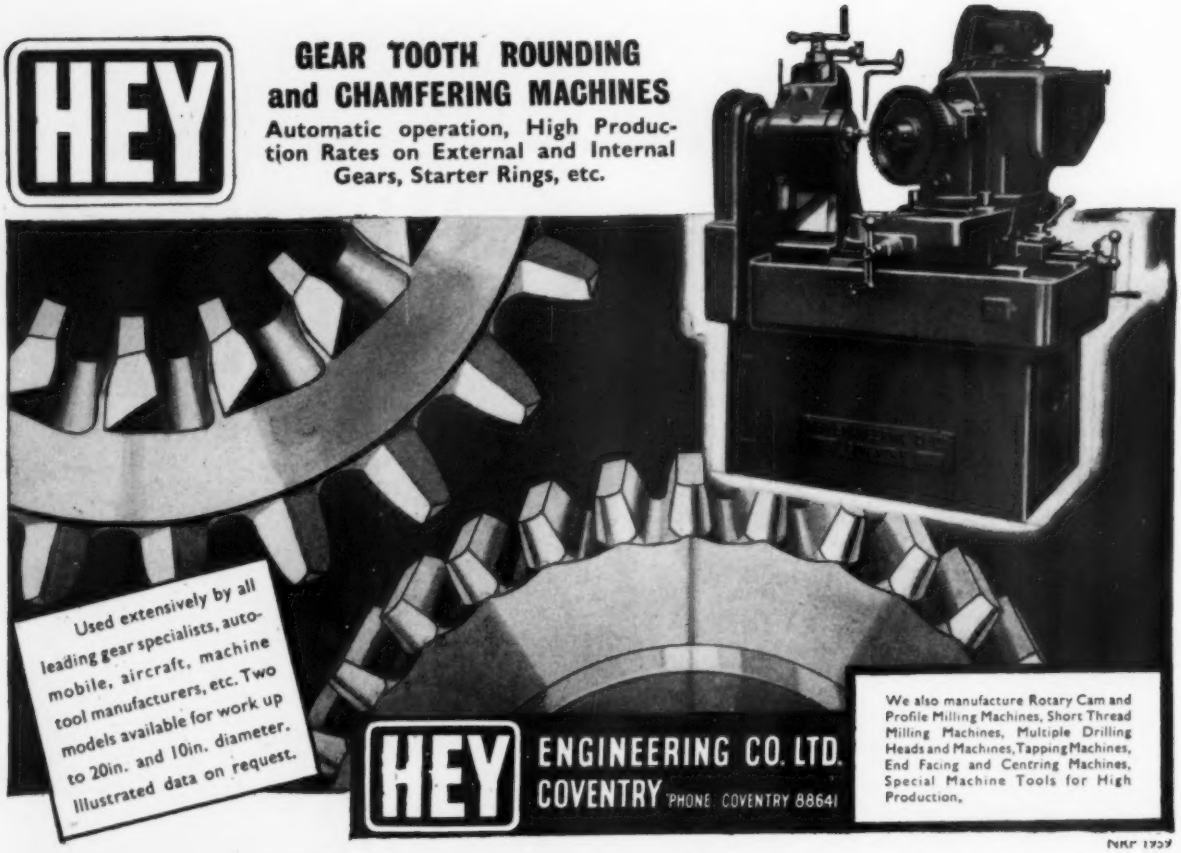

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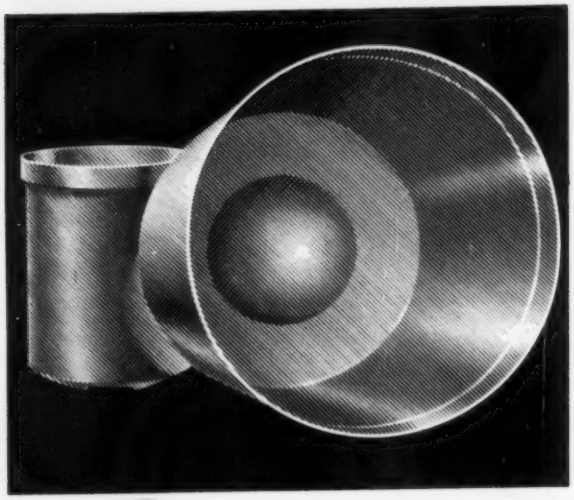

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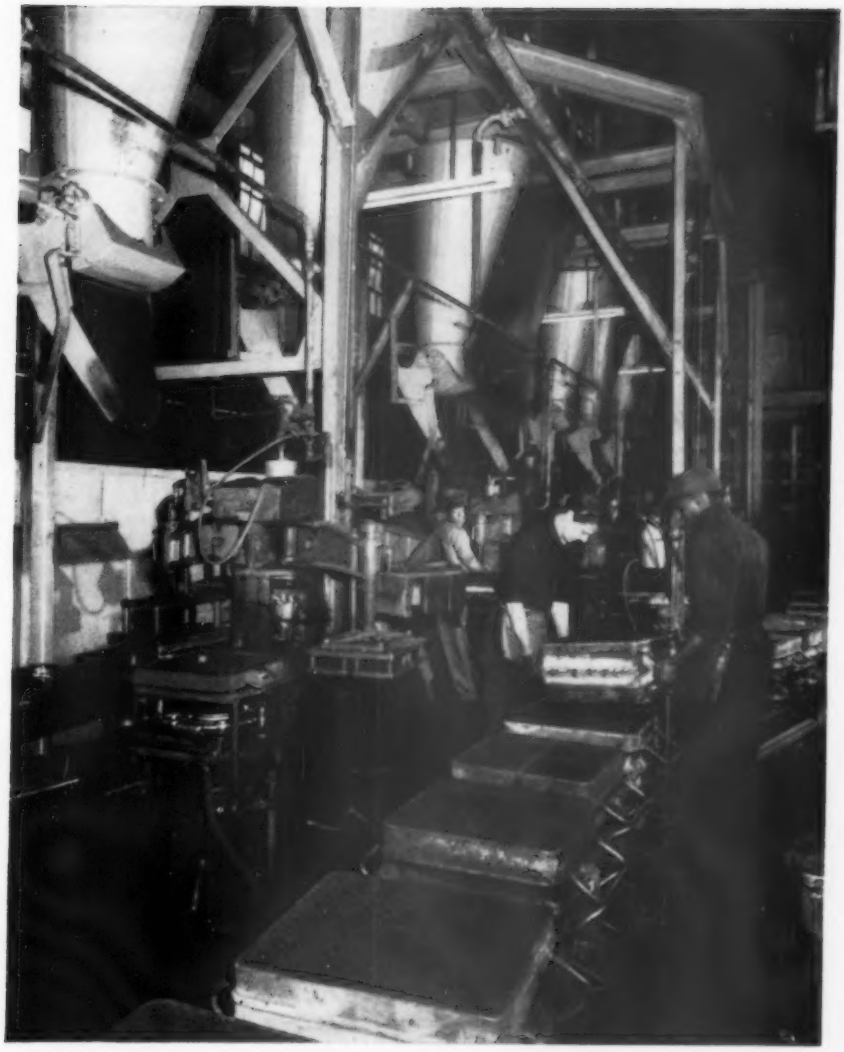

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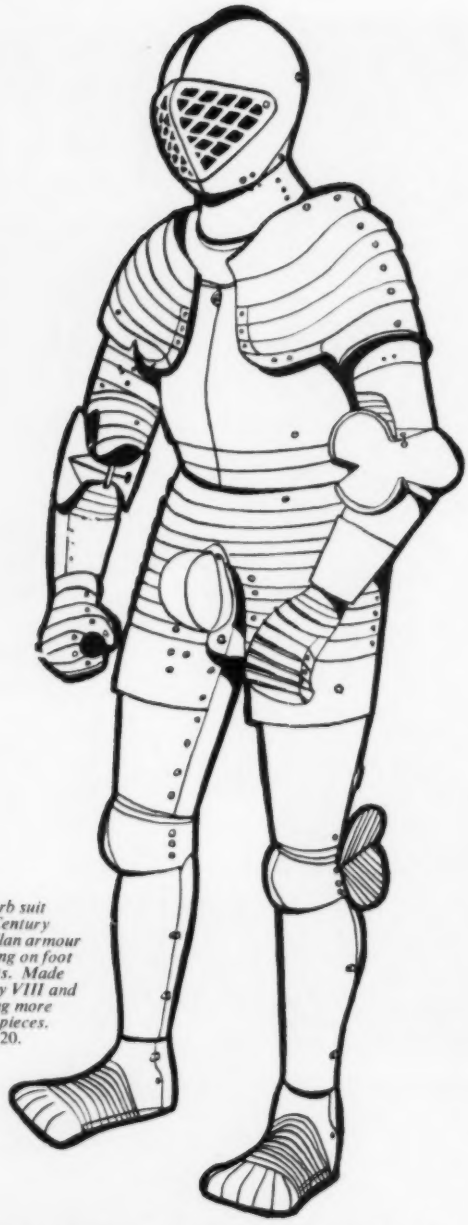

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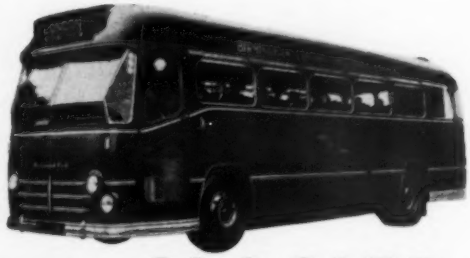

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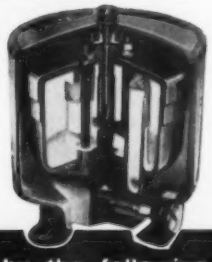

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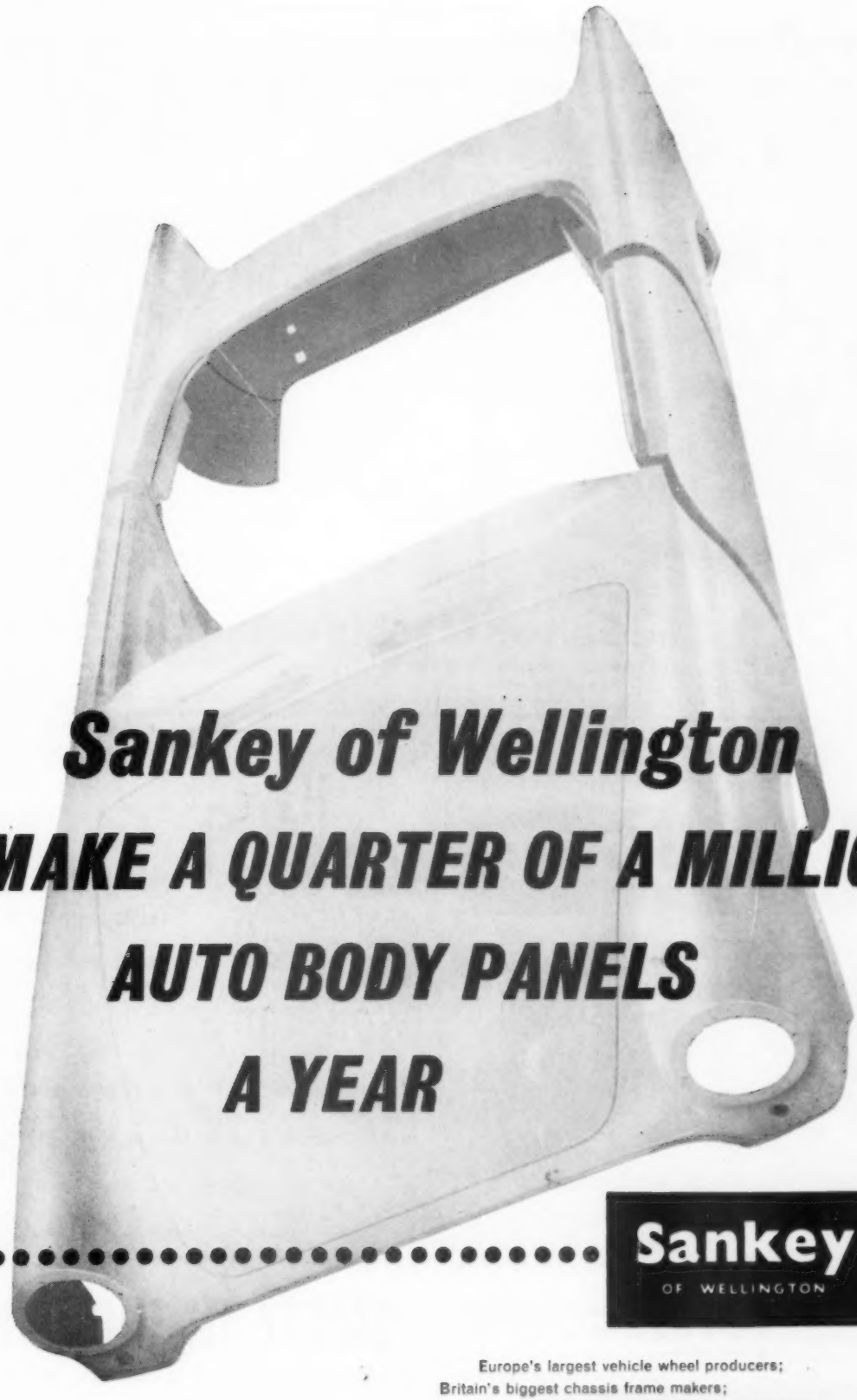

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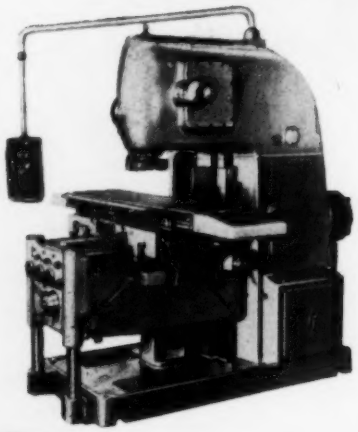

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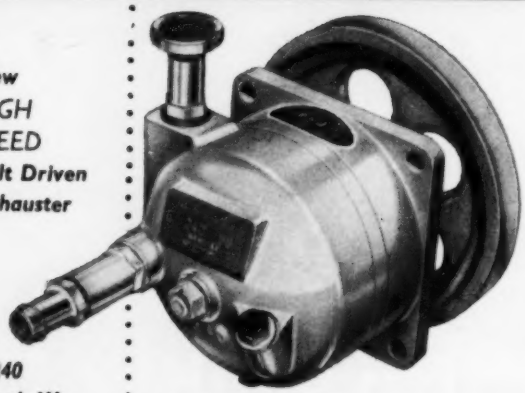

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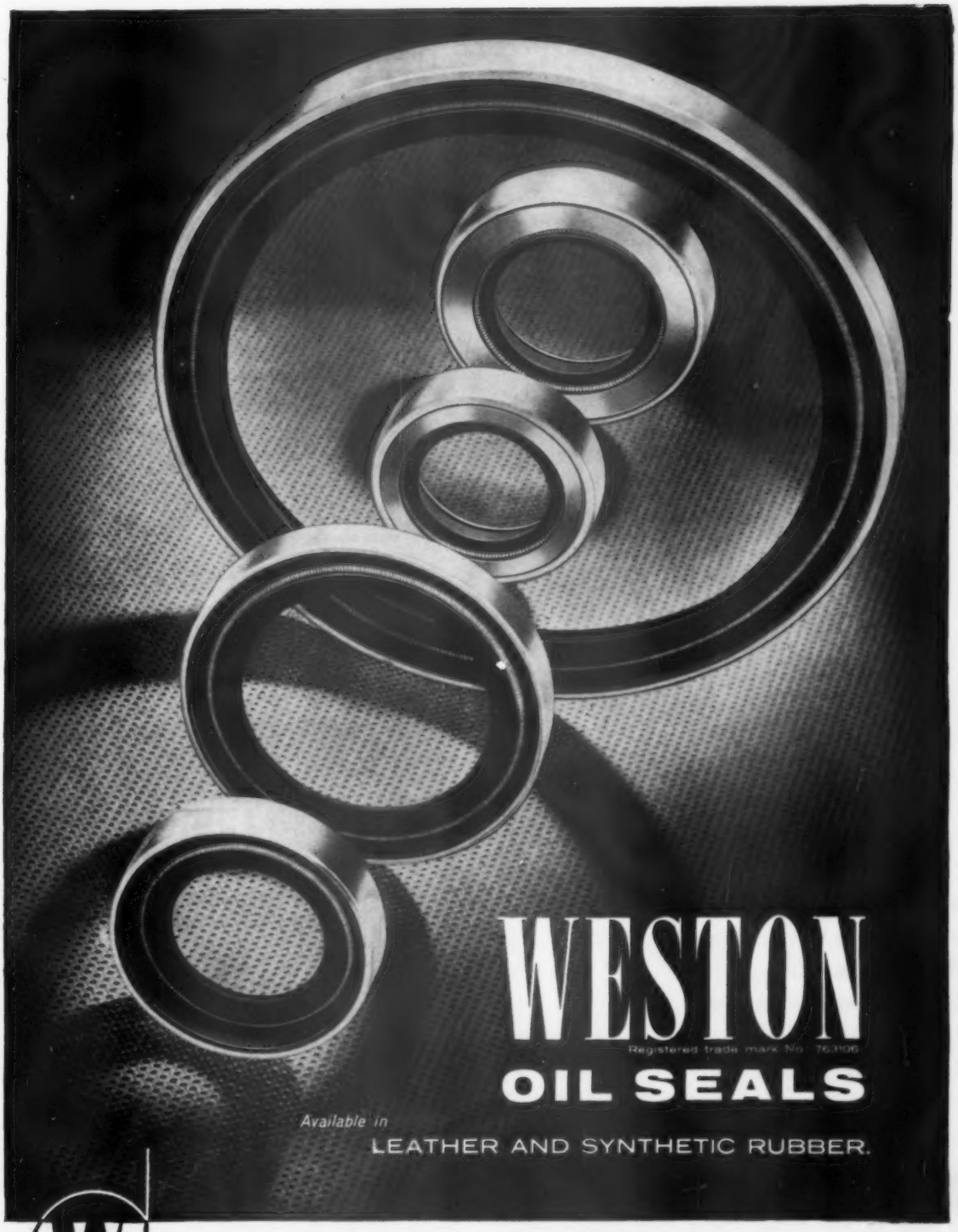

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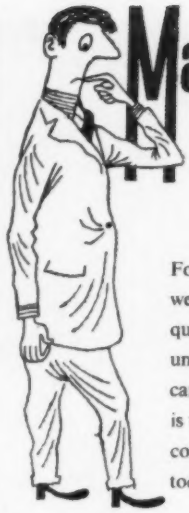

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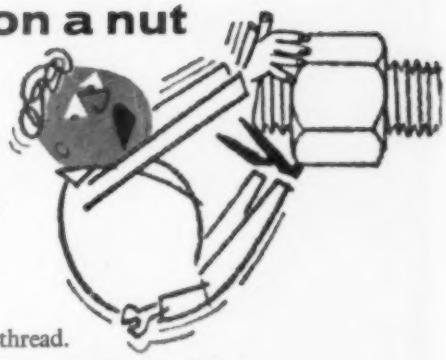

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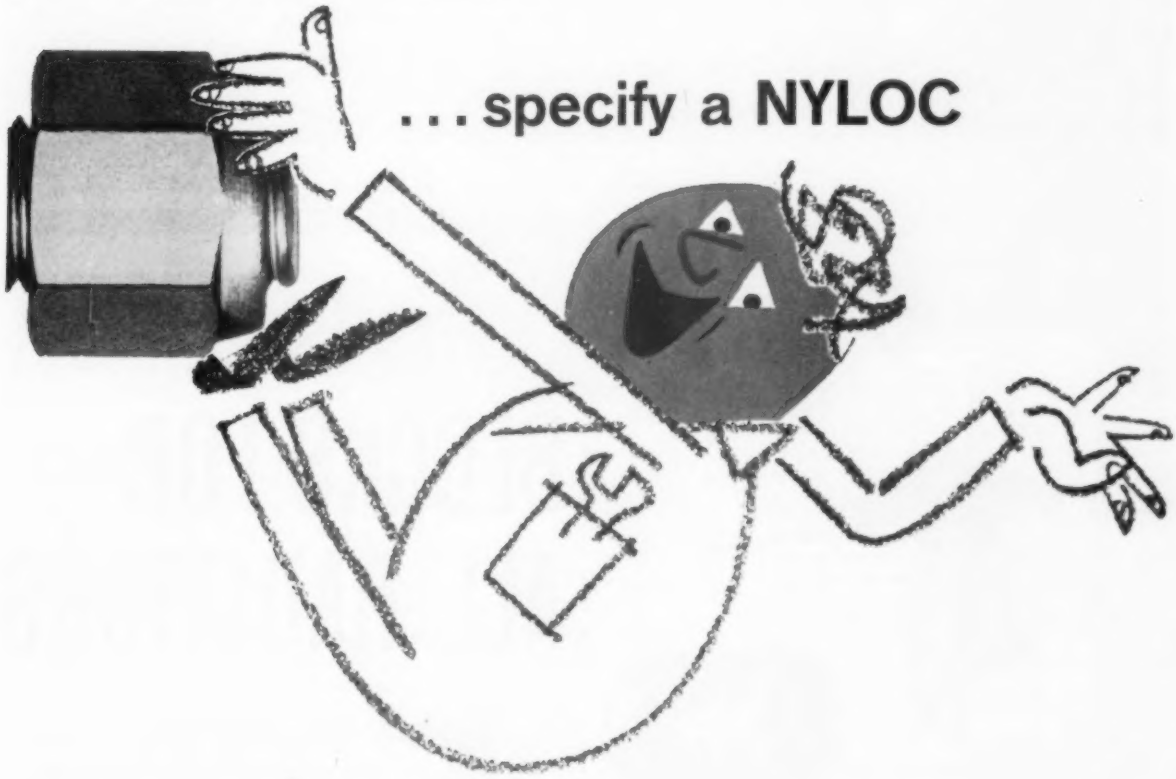

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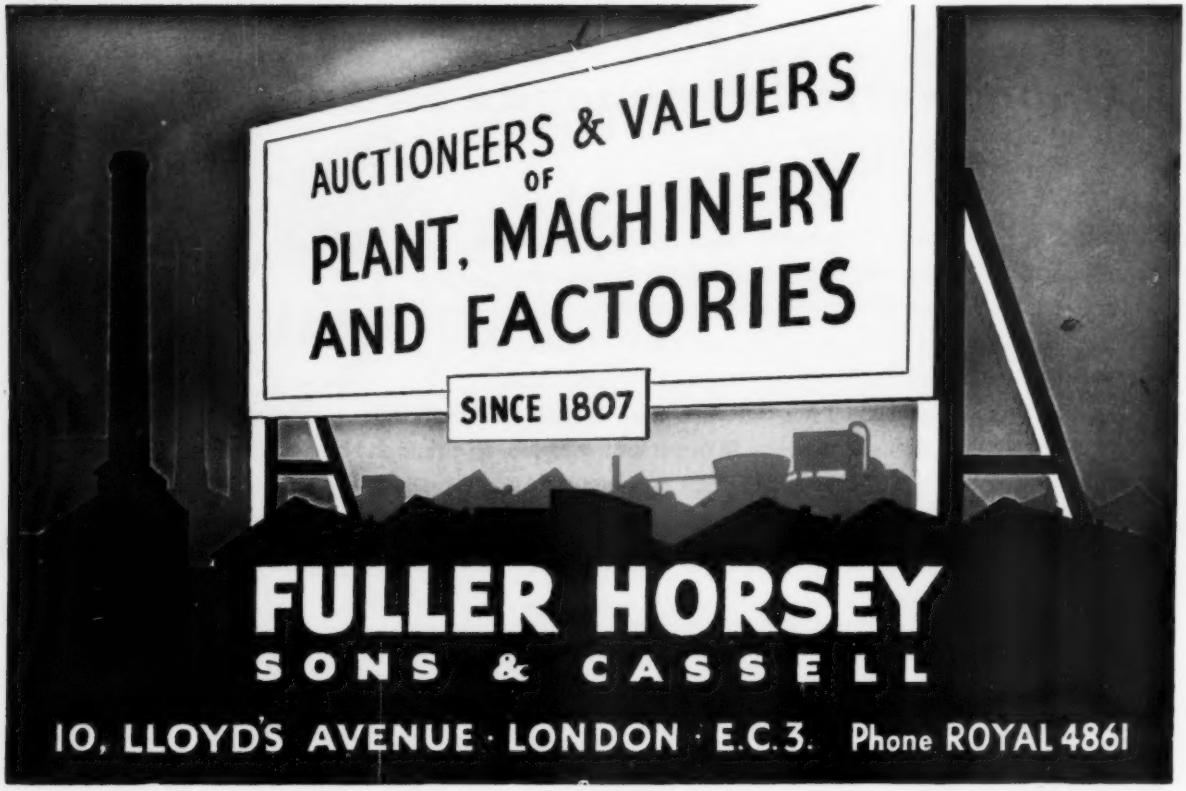

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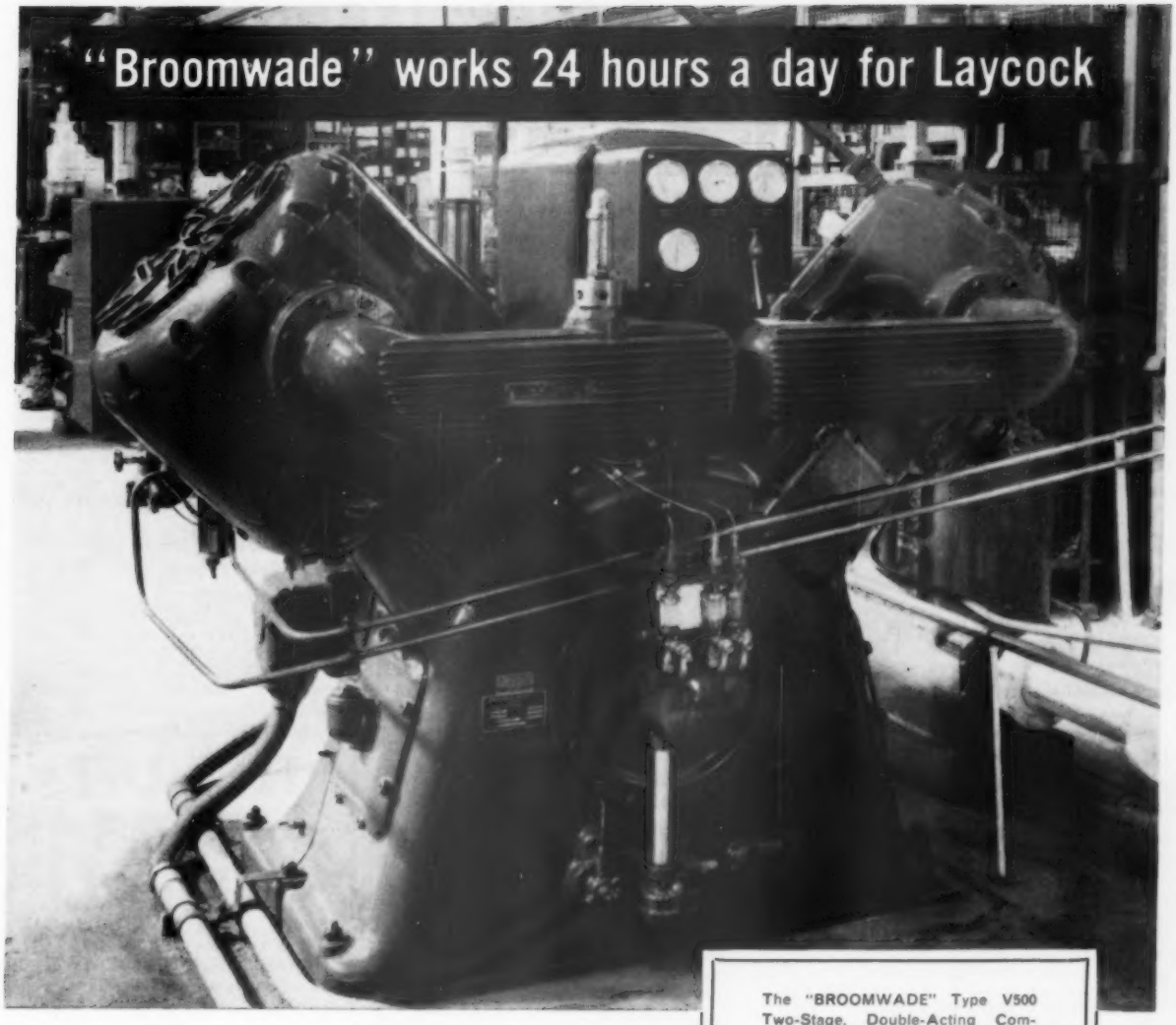

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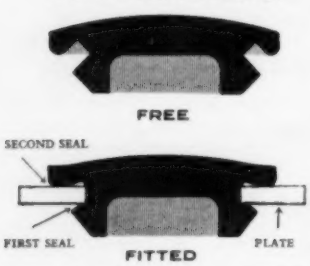

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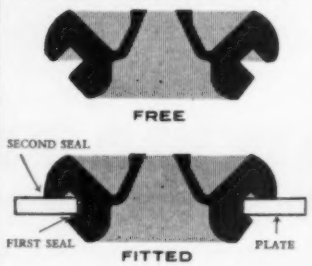

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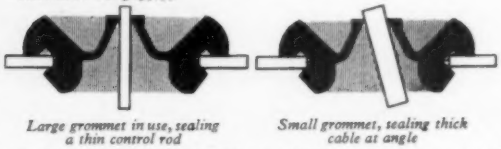

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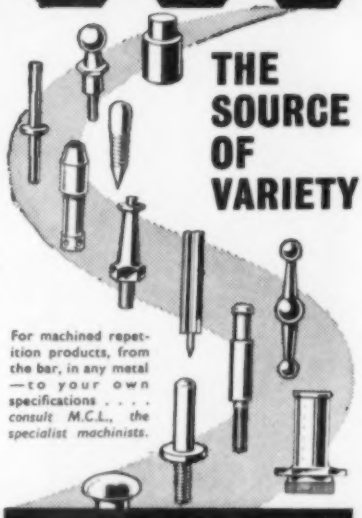

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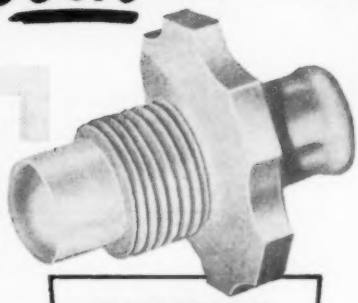

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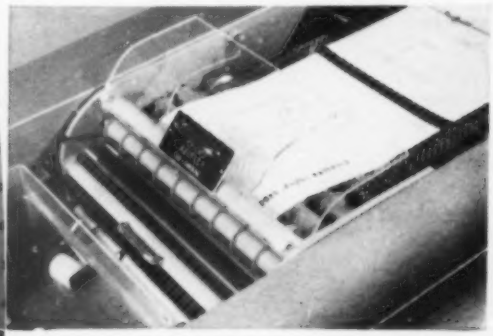

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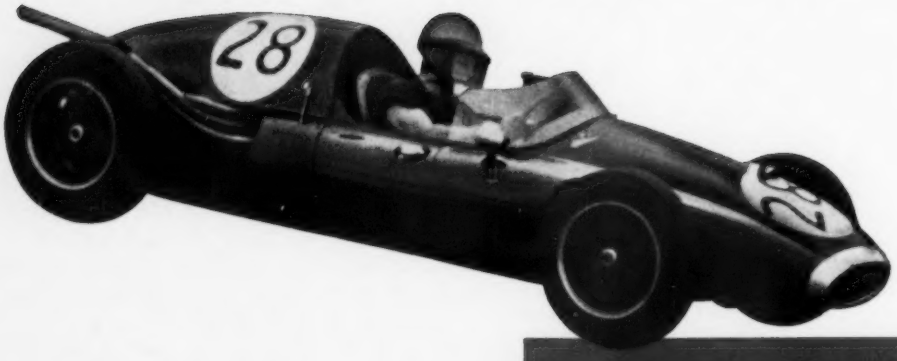

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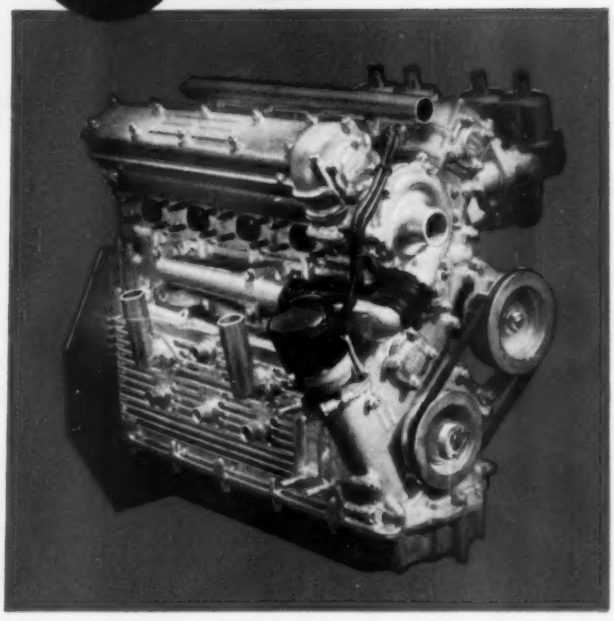

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